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Use of Seawater for Air Conditioning at Waikiki Convention Center

by Mike Williams

submitted to partially fulfill

the requirements of M.S. in Ocean Engineering

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Acknowledgments

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Synopsis

A large part of operating costs of a hotel in Hawaii is the cost of energy for air conditioning. Buildings can be constructed to use energy more efficiently by using many methods, however, some of these methods conflict with other concerns, aesthetics for example. Thus the process of designing and building an energy efficient hotel often involves trade-offs between energy efficiency and other objectives.

The method proposed herein to reduce energy costs is to introduce seawater, pumped from the deep ocean at a temperature of approximately six degrees celsius, directly to heat exchangers which cool the chilled water circulating in the building air conditioning system. The energy required to run the system would be reduced to only the cost of the seawater pumps, the fans and controls. The savings would be in the operating costs of the seawater pumps versus the cost to the compressors of a conventional air conditioning system.

One project which could conceivably employ such a system is the proposed future convention center in Waikiki. This paper presents a site specific conceptual design based on the proposed location of that project.

The proposed system would employ a high density polyethylene pipe of more than a meter diameter to deliver enough cold water to the site to accommodate the air conditioning load. The pipeline would have to be approximately 7.2 kilometers long and extend 5.6 kilometers offshore. The pipeline can be built with current technology.

Similar pipelines are operating at Keahole Point on the island of Hawaii, but this design would be the longest attempted to date. Current technology also exists to construct heat exchangers of either titanium or zinc-clad aluminum. It is likely that these could be ordered "off the shelf". Bio-fouling would not be a problem for either the pipeline or the heat exchangers as long as the cold water is not exposed to sunlight.

A seawater air conditioning system would be expensive to construct. However, the initial capital cost would be offset by the savings in electricity costs over the course of the first seven to nine years. Over the life of the system, the savings would be well into the tens of millions of dollars.

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Introduction

GENERAL INTRODUCTION

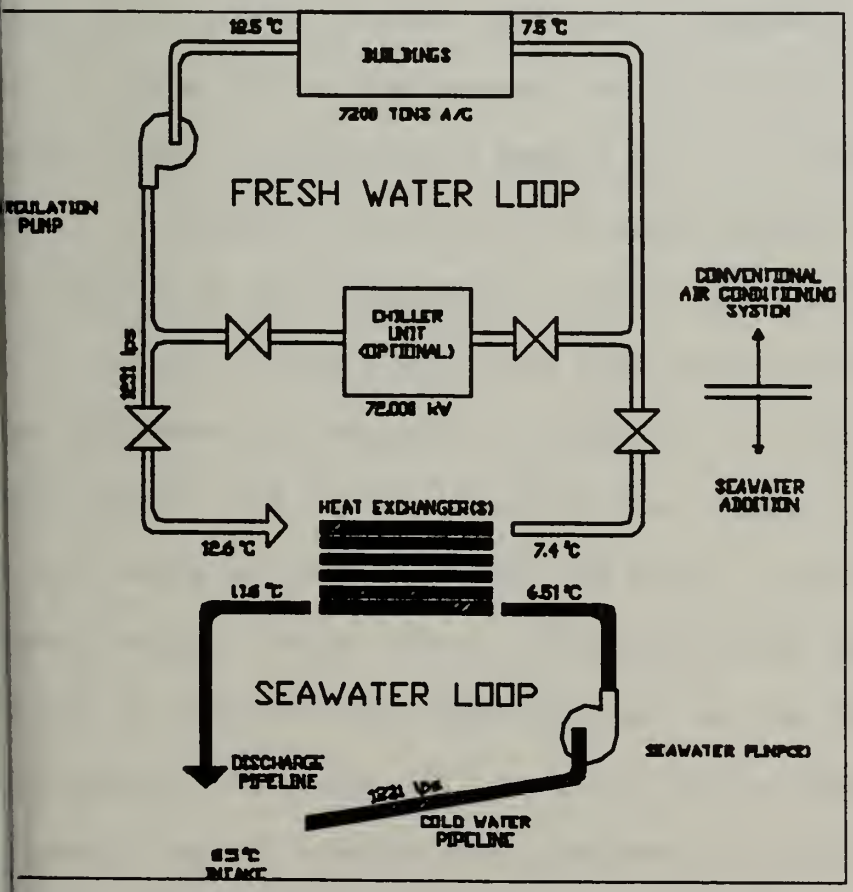
Air conditioning is very important to buildings in Hawaii. Hotels, shopping malls, restaurants and office buildings all rely on air conditioning to keep their guests, customers, patrons and workers comfortable. This is particularly important in the tourist industry where hotel operators want to keep their guests happy so that they might recommend Hawaii to their friends or return themselves.

Because of the importance of air conditioning, Hawaii business owners are willing to pay a high price to air condition their facilities, and they do. Air conditioning is expensive to operate, further driving up the costs of goods and services in Hawaii. Most of the operating costs are associated with the tremendous power consumption of air conditioning systems. About 1 kW of electricity is required to provide 1 ton of air conditioning in a conventional air conditioning system. The cost of electricity in Hawaii is high; on Oahu the cost is about $.10/kWh. There are well over 20,000 hotel rooms in Waikiki alone, each requiring approximately 1 ton capacity of air conditioning. This could amount to over $48,000 per day just for hotel room air conditioning.

Alternate technologies are available to help reduce the cost of air conditioning. The natural wind patterns could be used in some applications if buildings are designed to take advantage of them. However, this is not too attractive to hotels, for example, where

isitors expect reliable climate control. It is also impractical to etrofit existing buildings for natural wind usage.

Another way to air condition buildings without consuming a large mount of electrical power is through the use of the naturally cold ater which can be pumped up from the deep ocean. Such a system does ot need large, power consuming condensers, and would not use nlorofluorocarbon products which are blamed, in part, for the deple- ion of the ozone layer. The few buildings using the proposed deep eawater systems operate their air conditioning for approximately 20% f the cost of conventional systems.



g. 1 - Seawater Air Conditioning System dapted from Van Ryzin, 1991)

The idea behind such a system is that cold seawater pumped from the deep ocean is used to cool the water circulating in the air conditioning system through the use of heat exchangers (see Figure (1)). The heat exchange between the two fluids replaces the operation of the condensers or chillers in a conventional air conditioning

ystem. This process could be used to completely replace the condens-
rs, or could be used in conjunction with the condensers if either
ne seawater is not cold enough to provide 100% of the cooling re-
uired at peak times, or if a back-up system to the seawater system
s desired. This paper is intended to present a site specific concep-
ual design for Waikiki, Hawaii.

HISTORY OF SEAWATER AIR CONDITIONING

The idea of using natural sources of cold water for air condi-
ioning is not new. It has been appearing in literature for over
ixty years (Ciani 1980). Over the years, a variety of studies have
nown that energy savings from use of seawater or lake water can re-
ult in from 70% to 90% energy savings in places where there is a
rge, highly concentrated demand for air conditioning. The rows of
otels in tropical sites such as Miami Beach and Honolulu are sited
studies as good candidates (Ciani 1980; Hirshman and Kirklin
77). Less obvious sites have also been studied by the Naval Facili-
es Engineering Command. In addition to sites such as Naval Shipyard
arl Harbor and Pacific Missile Test Center in Point Mugu, CA, sites
far north as Cutler, Maine and Naval Security Group Activity in
nter Harbor, Maine (Ciani 1980) have also been looked at for feasi-
lity of seawater air conditioning. In the case of Winter Harbor, a
eliminary design in 1977 resulted in the conclusion that modifying
e existing air conditioning system to utilize seawater was the most
onomical alternative to more conventional upgrades.

In all the Navy investigated 241 sites in the 1970's for potentially economic use of seawater for air conditioning. At least twenty five of these sites were identified as having potential. Ten of those were identified as having a high potential. The sum of the energy savings at these ten sites was estimated at 23,000 MWh, at a cost at that time of $1.3 M per year(Ciani 1980).

A 1977 study by the Energy Research and Development Administration, ERDA, included a preliminary design of a large Miami Beach seawater air conditioning system. That study estimated that a 20,000 ton system would cost from $20.4 to $23.4 M, and would include a 2,000 foot (6.7 km) cold water supply pipeline (Roach, 1976). The cost estimates cited were admittedly crude at the time and these studies were based on 1977 prices. Similar construction would probably cost considerably more in 1994 dollars, however, construction materials and techniques for the costliest item, the cold water pipeline, have changed over the years. Many of the previous studies were based on steel pipe, whereas more recent pipeline installations have used polyethylene pipe. Despite the newer technologies, the cold water supply pipes still represent the largest portion of the capital costs of a seawater air conditioning system. The same study also concluded that cold water from fresh water bodies such as the Great Lakes, other large lakes and ground water from the northern third of the United States are also good sources for air conditioning systems (Hirshman, Kirklin 1977).

In recent years, the idea of using seawater for air conditioning as gone beyond the study and preliminary design stage. One system urrently using seawater to cool buildings is in Nova Scotia, Canada. n this system, water is pumped out of Halifax Harbor. Although Nova cotia does not have the cooling requirements of Hawaii, the owners f that complex are very satisfied with the performance and speed at nich the capital investment is being paid off; they plan to cool fu- ure expansions of their facility in the same manner (Tenbruggemcate 993).

At the Natural Energy Laboratory of Hawaii, another such system s also currently in operation, and cools the two main buildings at nat site. This system uses cold ocean water from an approximate epth of 600 meters, supplied by means of a high density polyethylene ipe.

The Natural Energy Laboratory of Hawaii (NELH) is located at Kea- ole Point on the west coast of the Island of Hawaii. There, a 30 cm iameter deep seawater pipeline has been in operation since 1982. ince 1987, four other pipelines have been added at the same loca- ion. Two 38 cm pipelines were deployed by a private firm. The state Hawaii deployed another 45 cm pipeline for "back-up" and a 1 m di- meter pipeline now supplies cold water to both the State of Hawaii)ean Science and Technology Park and the Ocean Thermal Energy Conver- ion (OTEC) research project at NELH, which is sponsored in part by e U.S. Department of Energy (DOE). The pipelines supply cold water r a variety of purposes, including aquaculture, OTEC research, pro-

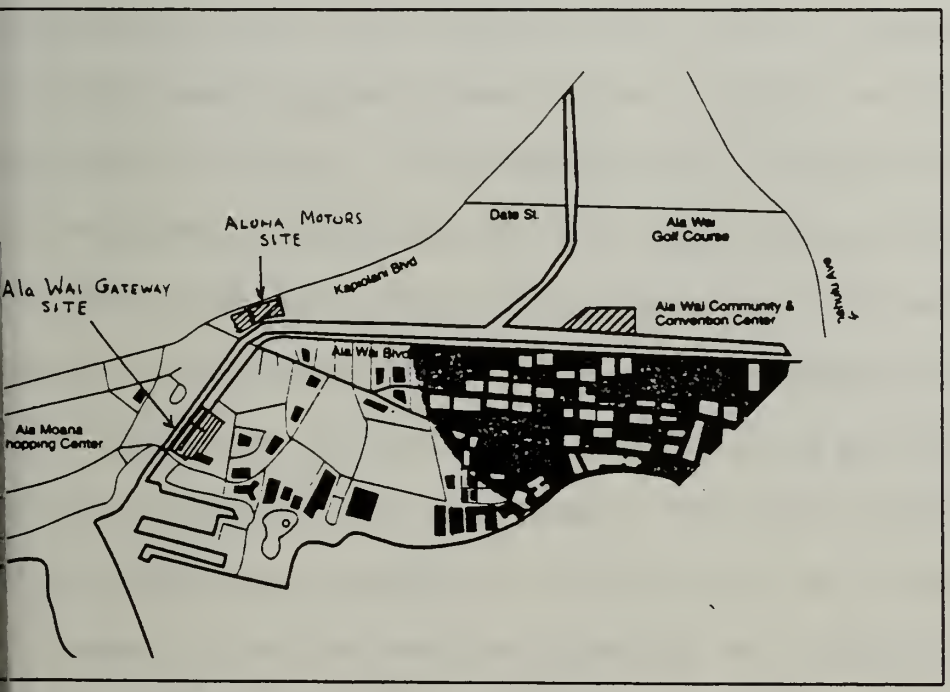
ıction of fresh water and the air conditioning of the laboratory ıildings (Daniel 1989).

All of the pipelines mentioned above are constructed of high den-ıty polyethylene. A variety of design options have been used to suc-essfully deploy and anchor these pipelines. The techniques which ıve proven successful there will be useful in future projects.

In this conceptual design, the operating temperature of the cold ıter can be assumed to be similar to those of the system which has een successful off Keahole Point, Hawaii. The load on the air condi-ıoning system is based on a rough estimate of the air conditioning oad of a future convention center in Waikiki, which has yet to be de-ıgned. The amount of water supplied will be based on that load. The awater will cool the fresh "chilled" water, which will circulate in e air conditioning system. The latent heat is included in the gross timate of the air conditioning load at this stage. Future design ases would have to consider the system in much more detail.

DESCRIPTION OF CONVENTION CENTER PROJECT

This conceptual design is intended to be site specific. The site hosen is one of three possible sites for a future convention center roject in Waikiki. Currently, the convention center project is only n the preliminary stages of planning. There were three sites in Waikiki which were under considera-



g. 2 - Map of Waikiki with Three Proposed onvention Center Sites

tion (see Figure (2)). One of the sites is at the old Aloha Motors site, the second is on the Ewa end of the Ala Wai Golf Course, and the third is across the Ala Wai from the Aloha Mo-

rs site, closer to the Ala Wai entrance. Design details are nonex-tent. Nevertheless, enough information is known to present a concep-al design for an air conditioning system which would use seawater r cooling.

As might be expected, the larger the expected air conditioning load, the easier it is to justify the capital expense of constructing a cold water pipe from Waikiki to water depths capable of use in an air conditioning system. For that reason, the convention center proposal containing the greatest need for a larger air conditioning system was selected for the basis of this concept design. That is the proposal to build a convention center and hotel complex on the old Aloha Motors site, as originally put forth by Sukamto Holding Corp. The other two proposed plans did not include any hotel rooms, and were smaller overall. The proposed hotel complex for the Aloha Motors site originally consisted of up to four towers, each up to four hundred feet high, with 2800 hotel rooms, but this has since been changed. In December of 1993, the state agreed to purchase the Aloha Motors site from Sukamto, and is expected to put the project out for separate proposals for development. This will probably result in a scaled down hotel portion of the facility (an estimate of only 500 hotel rooms is now included)and possibly an expanded or expandable convention facility.

DESCRIPTION OF PROJECT SITE

A design of the seawater air conditioning system for this project requires a cold water supply pipe of large enough diameter to deliver an adequate cold seawater supply at a temperature low enough to meet the cooling requirements of the convention center complex. Water at the required temperature, approximately 6°C, is generally found at

epths of 500 meters and greater. Such a pipe could, in concept, be aid in the Ala Wai Canal, adjacent to the Aloha Motors site and out hrough the Ala Wai Boat Harbor toward deeper water. There is a redged channel in front of the Hilton Hawaiian Village, but it may ot be practical to use it for a cold water pipe of the size and oute required.

Although, the water surrounding the Hawaiian islands is very eep, 1000 meters and more in places, as can be seen from the thymetry shown in Figure (3) and summarized in Figure (4), the ean floor off Waikiki slopes gradually for the first few kilome- ers. This gentle slope makes it necessary to construct a cold water pply pipe much longer and more expensive than those constructed at eahole Point. The area of the pipeline exposed to the surf zone uld also be larger than that of Keahole Point, increasing the diffi- lty associated with burial or securing the pipe to the bottom. This uld also increase the cost. These things must all be considered in th the design and in the economic analysis of such a project.

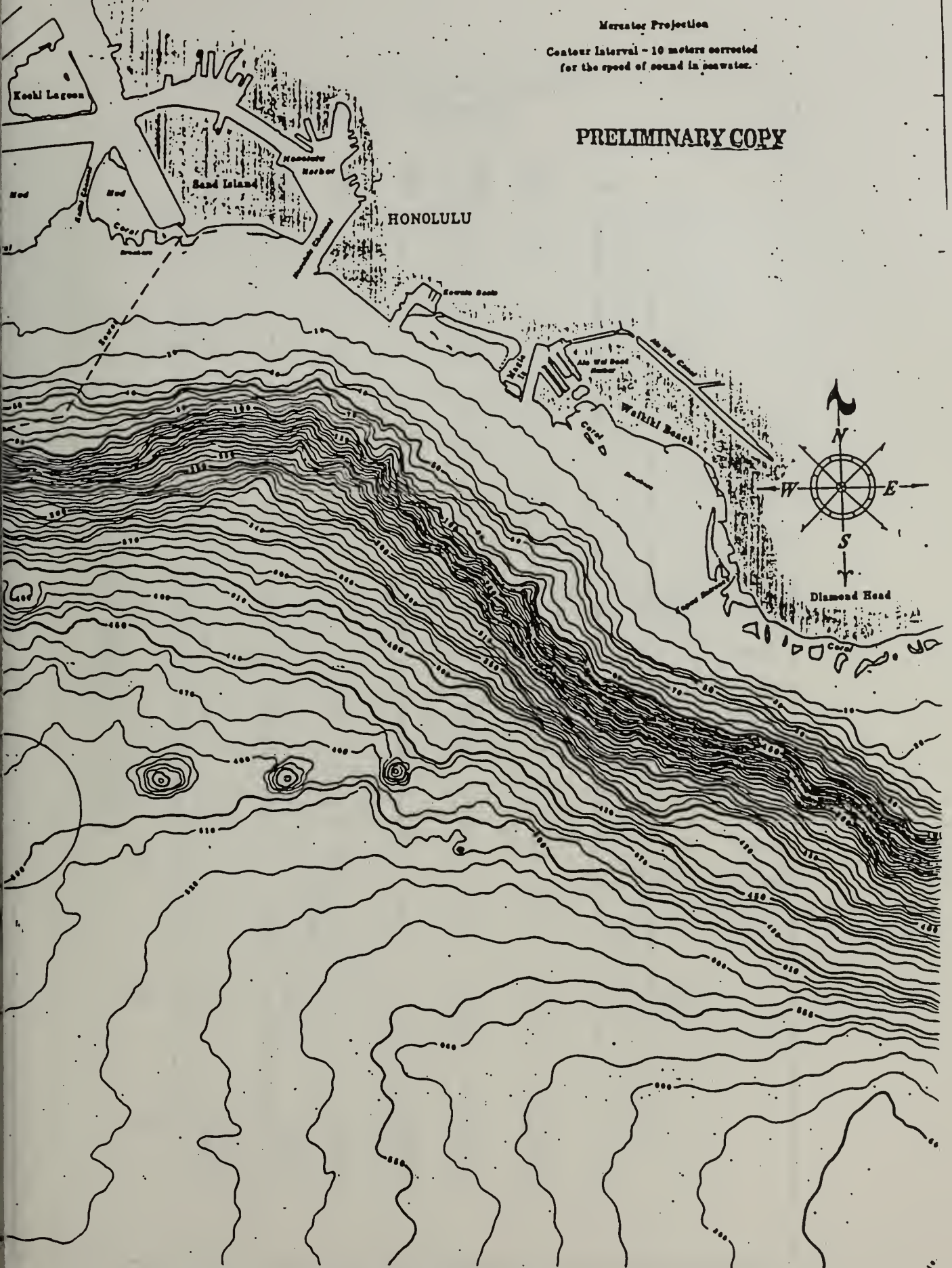
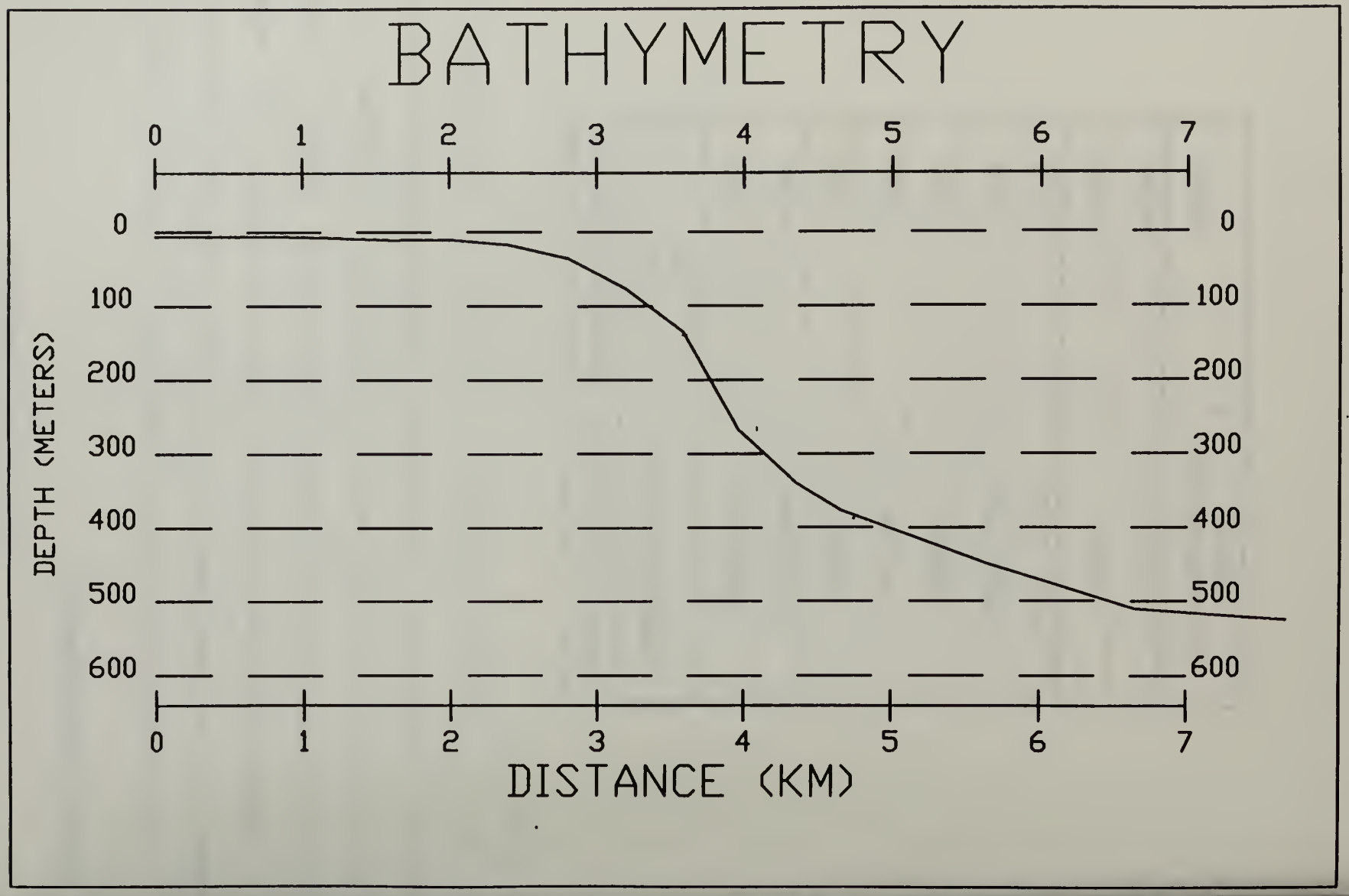


Fig. 3 - USGS Bathymetry Map

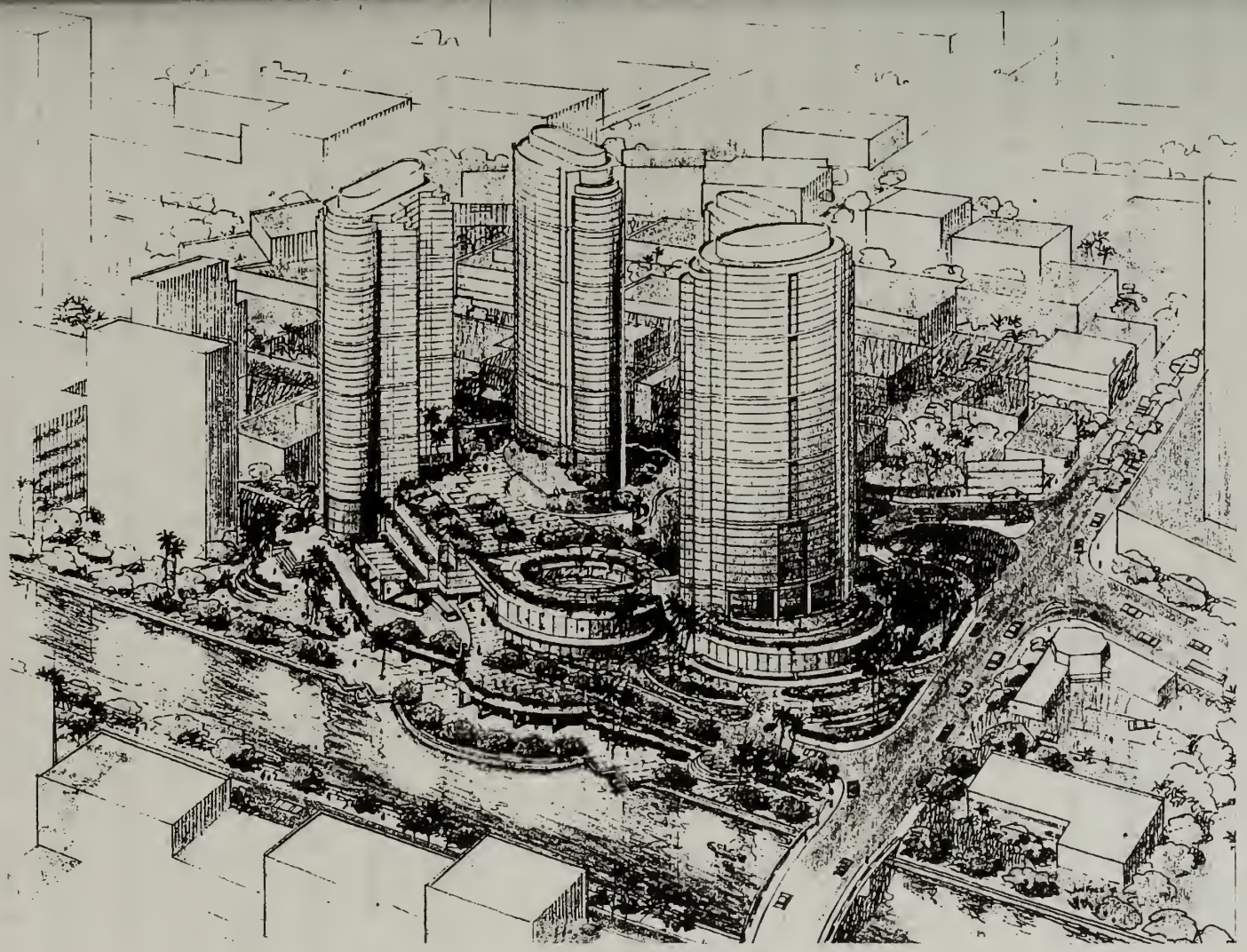

BATHYMETRY
0
1
2
3
4
5
6
7
0
100
200
300
400
500
600
DEPTH (METERS)
DISTANCE (KM)

COOLING REQUIREMENTS

The proposed Aloha Motors development is shown in Figure (5). A ummary of the significant building characteristics is shown in Table). Judging from the information available, however, this appears ily to include the first phase of the proposed complex. To comply ith state and city requirements, the complex is intended to have the apacity to expand in a second phase, increasing convention space to 0,000 s.f., meeting room space to 115,000 s.f. and ballroom space 65,000 s.f.

Table (1) Convention Center Size Estimate taken from Sukamto Executive Summary

CONVENTION CENTER	GSF (ft^2)
Exhibit Halls	200,000
Meeting Rooms	100,000
Ballroom	35,000
Pre-Function	75,000
Registration	25,000
Offices, Toilets	40,000
Service, Support	200,000
Sub Total	675,000
Parking	280,000
TOTAL CONVENTION CENTER	955,000



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HAWAII CONVENTION CENTER

Fig. 5

Also not included in Table (1) is the space for hotel rooms. The otal proposal originally included four towers, each up to 400 feet gh (40 stories) for a total of 2800 hotel rooms. This was a distin-ishing characteristic from the other two proposals being considered om the stand point of air conditioning load. Table (2) takes those ings into account and summarizes the building characteristics on ich the air conditioning load estimates for a seawater air condi-oning system could be based. The first two estimates in Table (2) flect the original Sukamto proposal. As stated earlier, however, it n be anticipated that the complex to be built may not have as many otel rooms as Sukamto originally proposed. To that end, a third esti-te of air conditioning load is shown in Table (2) with only 500 ho-l rooms and thus 500 tons of air conditioning projected. The other aracteristics will probably be about the same since they comply th the criteria set forth by the State.

The estimates of air conditioning load can get fairly compli-ted when building details are available. In this case, the building tails are sketchy at best, and the estimate of air conditioning re-irement is, therefore, admittedly rough. A "rule of thumb" that can employed for the hotel rooms is from .75 to 1.0 ton of air condi-oning per hotel room. When referring to cooling capacity, one ton capacity is equal to 3.5 kW or 200 BTU/min, and is defined as the ntinuous cooling rate provided over a period of twenty-four hours melting one short ton (906 kg) of ice. In the third estimate, one

n per room was used for simplicity, since the 500 room capacity is guess at best.

Table (2) Convention Center Air Conditioning Load Projections

CONVENTION CENTER	GSF (ft^2)	1ST ESTIMATE (1st rough guess) (tons)	2ND ESTIMATE (comparison to Blaisdell) (tons)	3RD ESTIMATE (combined after state land purchase) (tons)
Exhibit Halls	460,000		3382	3382
Meeting Rooms	115,000		1008	1008
Ballroom	65,000		448	448
Pre-Function	75,000		551	551
Registration	25,000		95	95
Offices, Toilets	40,000		151	151
Service, Support	200,000		1500	1500
Sub Total	980,000	3920	5634-7134	5634-7134
Hotel Space	700,000	2100-2800	2100-2800	500
Parking	280,000	0	0	0
Total Convention Center	1,960,000	6020-6720	7834-9934	6134-7634

The other areas of the complex are not covered by any such "rule thumb." The loads and ventilation requirements in the different eas vary widely. Exhibition halls full of people require a high ca- city for air conditioning per square foot, as do restaurants and chen areas. Ballrooms may also be either full of people or vacant. initial estimate can be based on a gross assumption that the over- requirement of 980,000 square feet requires approximately the e amount of air conditioning per square foot as the hotel room. s is the methodology applied in the first estimate in Table (2). A

tal load of 6720 tons can then be derived. A comparison to the her local convention center is made in the second estimate of Table). In this estimate, the sizes of the chillers for the various ar- s of the Blaisdell Exhibition Center, a much smaller facility, were ed. A direct per square foot relationship is assumed. The purpose some of the areas described in the Sukamto proposal is unclear, so assumption was made for the service areas that they may be either avy for that area, 1500 tons, or not air conditioned at all, such a loading dock area. Using this comparison, the expected load for e entire complex may be between 7834 and 9934 tons. The comparison the Blaisdell Exhibition Center indicates that the first gross es- mate may be a little low, but is at least of the same order of mag- tude.

Some additional factors may shed some light on this rough esti- te process, one being that the Blaisdell complex was built many ars ago when little attention was paid to energy conservation. Con- ruction technology has improved in that respect and a new conven- on center design would hopefully be more energy efficient.

Another factor is that the larger facility can be expected to be sier to air condition due to less wall area exposed to the elements square foot of building. This may be a big factor. For example, 400,000 s.f. building would have 2524 linear feet of wall (if it re a square). But, by comparison, a building with 1/10 the area, 000 s.f., would still have 800 linear feet of wall, 1/3 the amount the larger building. Of course, ratios of roof areas would theo-

etically be the same as the building square footage ratios, but the fficiency of the larger building is apparent.

Yet another factor is that it is unlikely that the ballrooms, exbition spaces, hotel rooms and restaurants would all be cooled at ace. It could be argued, therefore, that a conservative utilization actor of .75 should be used to multiply the total possible load in he second estimate, significantly reducing the expected load. This s less of a consideration in the third estimate since fewer hotel ooms would indicate that the facility would be used more purely as a nvention facility only. Taking the factors mentioned above into acount, a refined version of the estimates is shown as the third estite. The load used in the cold water requirements calculations is 00 tons. This is on the higher, more conservative side of the estite between 6134 and 7634 tons.

COLD WATER REQUIREMENTS

With an air conditioning load of 7200 tons, enough cold water st be supplied to the heat exchangers to remove 25,200 kW of ther- l energy from the fresh water which circulates in the air condition- g system. By definition, 1 ton equals 3.5 kW of thermal energy. om the experience of previous projects, a 5°C increase of the seawa- temperature in the heat exchangers can be assumed. Then the folving relation must be true,

$$C_p \cdot 5°C \cdot M = 25{,}200 \text{ kW} \qquad \text{Eq. (1)}$$

where:

M = mass of seawater through heat exchanger

C_p = specific heat of seawater $\cdot \frac{kJ}{kg\ °C}$

$= 3985 \frac{J}{kg°K}$

(See Appendix B-3 for specific heats of seawater at various temeratures and salinities.)

$$3985 \frac{J}{kg°C} \cdot 5°C \cdot M \left(\frac{kg}{s}\right) = 25{,}200\ kW \left(\frac{kJ}{s}\right) \qquad \text{Eq. (2)}$$

Solving for M:

$$M = 1265 \frac{kg}{s}$$

$$\text{Volumetric Flow } Q = \frac{M}{\rho} = \frac{1.265 \frac{kg}{s}}{1027.6 \frac{kg}{m^3}} = 1.231 \frac{m^3}{s} \qquad \text{Eq. (3)}$$

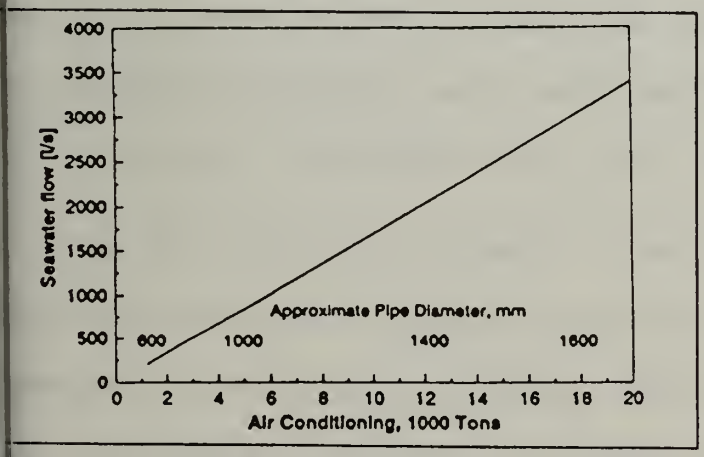


Fig. 6 - Seawater Flow (from Van Ryzin, 1991)

The flow must then be 1231 lps (19,510 gpm). When specifying a heat exchanger, some vendors are more receptive to gallons per minute. The 1231 lps value agrees fairly closely with Figure (6) which was previously determined (Van Ryzin 1991).

Knowing the volumetric flow requirement to achieve the desired cooling allows for design of the cold water pipe and seawater-freshwater heat exchanger.

MATERIAL

In many older designs for ocean pipelines the basic pipeline ma-
rial used in the petroleum industry, for example, is steel. Exter-
l and internal coatings were specified for corrosion protection and
draulic flow enhancement. Concrete jackets were specified to pro-
de added weight, thereby providing stability. This weight was con-
dered too great for pulling into place, and the pipeline sections
uld have had to be voided during the pulling operations. This was
expensive and difficult process. Newer technologies make things
sier.

Plastic pipe, now available in large diameters, has several im-
rtant advantages over steel pipe in this application. It will not
rrode in seawater and it has good thermal insulation properties.
o candidate materials are, high density polyethylene pipe (HDPE)
d fiberglass reinforced resin pipe (FRP).

FRP has excellent strength-to-weight characteristics. It can
thstand burial and external pressure. Its axial strength-to-weight
aracteristics permit extremely long sections to be bottom-pulled.
is resistant to seawater (minimum life of 40 years), has excellent
ermal insulation properties, and is available in specific gravities
nging from 1.6 to 2.08. This is an advantage where negative buoy-
cy is desired.

The high density polyethylene pipe is attractive because it can, some situations, be continuously extruded at the site. It is avail-le "off the shelf" in diameters up to 63 inches in 40 foot sec-ons. It has been successfully deployed in the ocean in the llowing manner. Its end is sealed, and it is extruded into quiet wa-rs. Concrete weights are bolted to the pipe at regular intervals ile it is afloat. These concrete anchors, once installed, do not rmit bottom pulling. When a sufficiently long section is assembled, is floated to the site and installed by controlled flooding. This chnique is good for protected waters. In the Waikiki design, the a Wai Canal could be the staging area.

Polyethylene, a type of plastic, is a polymer made up of long ain molecules. Polymers can be formed in nature or can be manufac-red by man from natural materials. Polyethylene falls into the cate-ry of plastics known as thermoplastics. These plastics soften when ated and can be reformed many times. Pipe made from polyethylene n be extruded in various lengths. Other parts such as fittings can formed by injection molding. Other thermoplastics include polyvi-l chloride (PVC), polypropylene (PP) and acrylonitrile butadiene rene.

Polyethylene's characteristics make it a good choice for a cold water supply pipeline for a number of reasons. Its resistance to rosion is excellent, giving it a great advantage over steel. It adequate stiffness and retains its cross section, yet it is very ht compared to steel, making it easier and cheaper to handle at

e site. It also is a better insulator than steel. This is an advan-
ge in areas where the cold water pipe is routed through shallower,
rmer water.

There are, however, drawbacks of HDPE. One major disadvantage
ich arises from polyethylene's lightness is that with a density of
960 to 0.965 g/cm^3, the pipe is buoyant even when flooded. It
erefore requires a great deal of effort to anchor the pipe in
ace. This must be addressed in the design. Another consideration is
at, as with many plastics, the creep modulus of HDPE is important.
is can affect the service life of the pipe if not factored into the
rength calculations (Rubin 1990). Another drawback is the suscepti-
lity to abrasion that could occur due to motion of the pipe while
contact with the sea floor.

The advantages of high density polyethylene (HDPE) appear to out-
igh the drawbacks, and therefore, it is the material chosen for the
ld water pipeline. This choice is made easier mainly due to the suc-
ssful experiments at Keahole Point.

PIPE LENGTH

The length of the cold water pipe is dependent on the bathymetry
shore of Waikiki and the depth at which the required water tempera-
re can be obtained. The required temperature of the intake water at
heat exchanger is expected to be in the vicinity of 6.5°C, from a
gh estimate based on the desired performance of the seawater -
shwater heat exchanger. From Figure (7), a temperature rise in the

ld water supply pipe of
ss than 0.25°C might be ex-
cted; however, Figure (7)
based on a study which em-
oyed a much shorter (220
pipeline (Van Ryzin
91). In light of this, an
itial estimate of 0.5°C
se in intake water tempera-
re can be used to establish

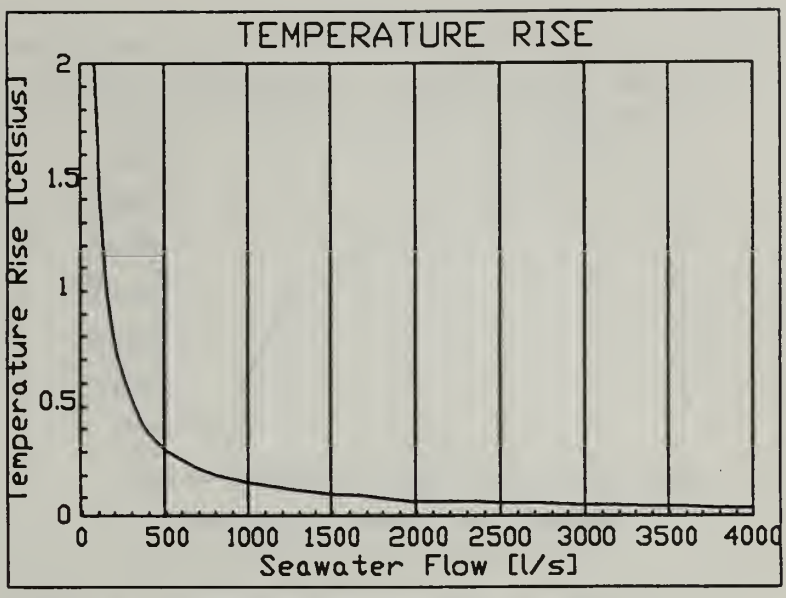


Fig. 7 - Temperature Rise

e pipeline length. Once the length is determined, a heat gain calcu-
tion may be performed for the length of the pipe and if necessary,
e pipe design length could be adjusted to a length and depth where
lder or warmer water could be obtained, whichever the case may dic-
:e. This iterative process could be performed several times to opti-
ze the design, however, the accuracy of the known conditions would
: warrant more than one or two iterations at this stage. A calcula-
on of heat gain is made to verify the reasonability of the 0.5°C
nperature gain assumption in Appendix A-3. This calculation pre-
:ts a temperature gain of only 0.1°C, but 0.5°C will be used as a
nservative estimate.

Since the desired heat exchanger seawater intake temperature is
°C and the assumed heat gain in the intake pipe is 0.5°C, the in-
of the seawater supply pipe should be positioned in a region
re 6°C seawater will be encountered. Figure (3), a yet unpublished

l preliminary U.S. Geologi-
. Survey (USGS) map, indi-
es the bathymetric
ntours as accurately as
y available source can at
s time. From a typical
water temperature pro-
le, Figure (8), a depth
be assumed at which 6°C
er may be obtained. The
th indicated by Figure (8) is approximately 525 m.

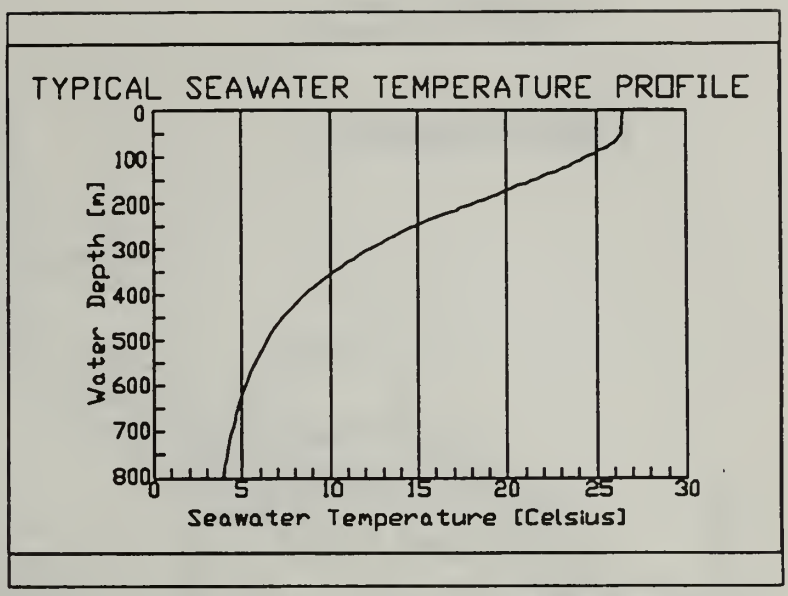


Fig. 8 - Typical Seawater Temperature Profile in Hawaii

Figure (9) indicates a pipeline route to 525 m depth using the GS map, Figure (3), as a reference. The shortest straight line dis-
nce to the desired depth is shown to be about 5.5 kilometers from
e Ala Wai Harbor entrance in a south by southwest direction.

The length of the pipe also needs to be adjusted for the depth
rtical component of distance) and any possible zig-zag required
route since bottom obstacles may be encountered and a straight
e pipeline is not a reasonable expectation. Large rocks, coral
ds and natural geological features will inevitably cause small de-
rs.

The vertical component of 500 meters is added to the horizontal
ponent of 5500 meters to get 5522 meters. In light of possible ob-
cles and bending of the pipe in suspended or catenary support con-

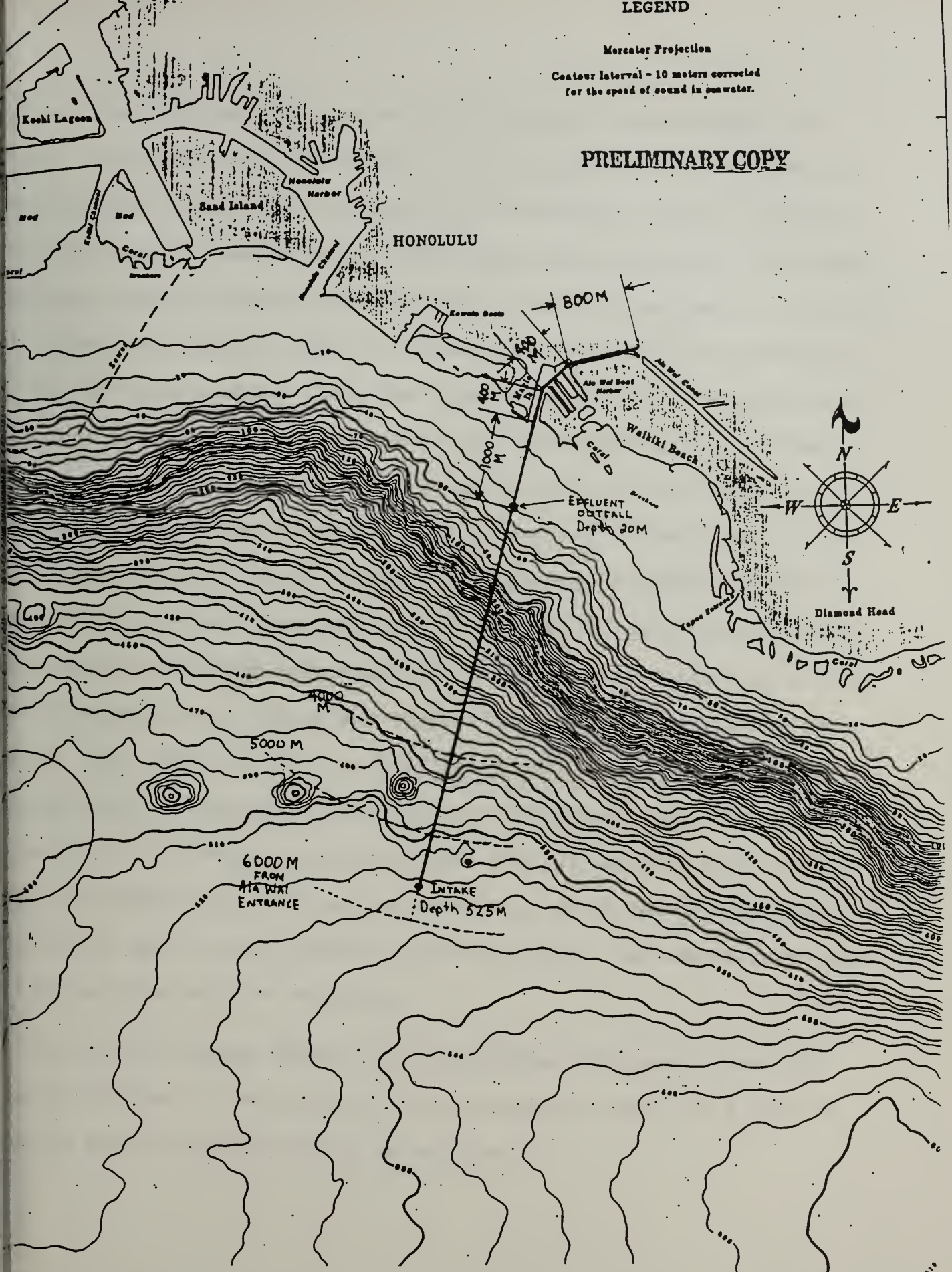


Fig. 9 - USGS Bathymetry Map With Pipe Route

gurations, 5600 meters is used as the distance from Ala Wai en-
ance. In addition to 5600 meters of pipe required for the distance
om the pipe inlet to the entrance of the Ala Wai Harbor, the pipe
st continue for approximately 400 meters into the harbor, then bend
ght and continue another 400 meters to reach the entrance of the ca-
l. After entering the canal, it continues about 600 to 800 meters
the convention center site. The pipe is exposed to a warm water en-
ronment throughout these sections. This becomes relevant in the
at gain calculations. The total length of pipe is 7200 meters for
is design, more than 3 times the length of any of the pipes at Kea-
le Point. However, this is a similar distance to that evaluated
om the Miami study (Hirshman and Kirklin 1977), with favorable eco-
nic results. The reasonableness of extending the intake pipe so far
ould be considered during any further design processes, since the
ital cost involved with the cold water pipe is so high. For the
culations of temperature gain in the cold water pipe, the total
stance of 7200 meters can be used for now. It will be shown by cal-
ations herein that the shallow areas are the areas in which the
perature gain is the greatest and even there, the temperature gain
ide the pipe will be very small.

For future design phases to be completed, the pipe route indi-
ed by Figure (9) would have to be surveyed in detail and the tem-
ature profile would have to be verified.

PIPE DIAMETER

Since the cold water requirement of 1231 liters/sec has been es-
blished, the inside diameter and flow velocity can be determined.
course, the faster the flow, the smaller diameter pipe can be used
d visa versa. Flow velocity also determines the pressure required.
o large of a negative pressure imposed on the cold water pipe could
llapse it, so care must be taken, after a velocity is picked, to de-
rmine the pumping requirements. It is advantageous to use as high a
locity as allowable, since the high velocity saves on pipe cost by
ducing the diameter and temperature gain due to losses to the out-
de seawater. Conversely, a slower velocity requires less pumping en-
gy because of less head loss. The final design should seek an
timum between pumping costs and pipeline costs.

Previous research indicates that a good starting point, as Fig-
e (6) shows, is an 1150 mm diameter pipe. A pipeline of a slightly
ller diameter has been deployed successfully off Keahole Point, Ha-
i, and is well within the state of the art (Van Ryzin, 1991). For
mple, with a diameter of 1150 mm, the calculation of velocity is
follows:

$$\text{Volumetric Flow} = \frac{\pi D^2 V}{4} = 1.3 \frac{m^3}{s} \quad (1300 \text{ lps}) \qquad \text{Eq. (4)}$$

where:

D = pipe diameter (inside) = 1.15m

V = flow velocity

This calculation yields a flow velocity of 1.25 m/s. With an es-
mate of inside diameter and velocity determined, the length of the
pe must be determined to establish pressure and temperature losses.
is conceptual design will be based on a 1200 mm pipe which has a
ow velocity of 1.329 m/s, as explained in Appendix A-1. A more re-
ned selection of pipe size can be made when the flow charac-
ristics are better defined.

PIPE FLOW CHARACTERISTICS

Understanding the characteristics of the seawater flow both in-
de and outside the cold water pipe is important in determining the
essure drop and head loss as well as the heat transfer charac-
ristics of the system. One objective is to determine the pressure
ss, and hence, the power requirements for the intake pumps to
nieve the desired flow rate for the inside of the pipe. Some other
sumptions are required to make calculations regarding the flow
rough a pipe. One is that the flow is incompressible so as to sat-
y equation (5), the continuity equation. Another is that the inter-
flow is completely bounded. This means that the flow velocity is
same over the entire length of pipe,

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad \text{Eq. (5)}$$

where:

u, v, w, are velocities in the x, y, z directions,
spectively.

In the case
pipe flow, v
d w are assumed
equal zero.

Figure (10)
a diagram of
e model of the
id water pipe
stem.

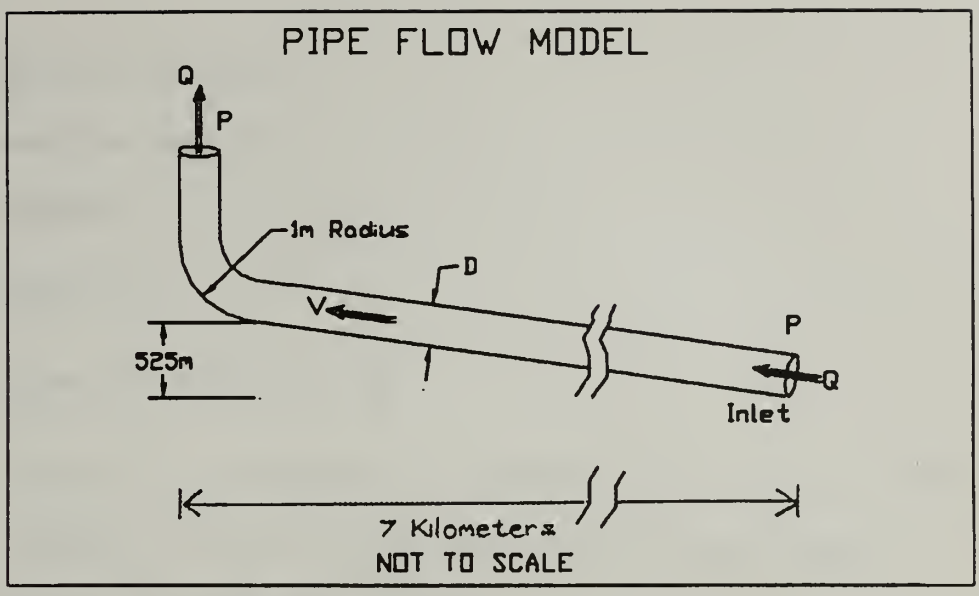


Fig. 10 - Pipe Flow Model

Head loss
er a length of the pipe can be expressed in the following equation:

$$\frac{P_1-P_2}{\rho} = g\,(Z_2 - Z_1) + h_{l_T} \qquad \text{Eq. (6)}$$

where:

h_{l_T} = Total head loss

P_1 = Initial pressure

P_2 = end pressure

Z_1 = Initial elevation

Z_2 = Final elevation

head losses must be calculated differently depending on whether
not flow is laminar or turbulent. Pipe flow may be either laminar
turbulent depending on the Reynolds number,

$$R_e = \frac{\rho V D}{\mu} \quad \text{or} \quad \frac{VD}{\nu}$$

where:

ρ = density of fluid

$\bar{V}$ = average velocity

D = diameter

μ = viscosity

ν = dynamic viscosity = $\frac{\mu}{\rho}$

$R_e \leq 2300$ then the flow is assumed to be laminar. $R_e > 2300$ is as-
med be turbulent. However, laminar flows may exist at much high
ynolds numbers under some conditions.

The change in pressure for laminar flow is computed analytically
th the following equation:

$$\Delta P = \frac{128\mu LQ}{\pi D^4} = \frac{32L}{D} \cdot \frac{\mu \bar{V}}{D} \qquad \text{Eq. (7)}$$

The head loss can be computed by:

$$h_l = \frac{32L}{D} \frac{\mu \bar{V}}{\rho D} - \frac{g(Z_2 - Z_1)}{\rho} \qquad \text{Eq. (8)}$$

It can be seen after some algebra that:

$$\frac{32L}{D} \frac{\mu \bar{V}}{\rho D} = \frac{64}{R_e} \frac{L}{D} \frac{V^2}{2} \qquad \text{Eq. (9)}$$

where for laminar flow $\frac{64}{R_e}$ is known as the friction factor.

s shows that for laminar flow, the pressure loss in the pipe is de-
ient on Reynolds number only, not on the roughness of the pipe
face.

For fully developed turbulent flow, the friction factor is deter-
ned experimentally. The pressure drop in this case is known to de-
nd on the Reynolds number and the pipe roughness:

$$h_l = f \frac{L}{D} \frac{\bar{V}^2}{2} \qquad \text{Eq. (10)}$$

where f is the friction factor

The relative roughness, e/D, must be determined from the pipe
aracteristics (see Appendix B-1). The friction factor can then be
termined from the exponentially determined curves in Appendix B-2.

Head loss represents energy converted by frictional effects from
chanical to thermal energy. Head losses result from what are termed
or and minor losses. Major losses include losses due to friction
er the length of the pipe. This is the main concern with a pipe 7.2
long. Minor losses are losses due to valves, tees, elbows and en-
nces. These are not considered here since the tees and elbows have
yet been determined.

Since the pipe diameter (inside) will be shown to be 1.086 m,
w velocity is expected to be 1.329 m/s, dynamic viscosity 1.56 x
^{6} m^2/s, and Re = 9.252×10^5. Therefore, the flow inside the pipe
be considered turbulent, and the losses are determined by
(10). Since it have been established that turbulent flow is the
e for the cold water pipe, Appendices B-1 and B-2 should be used.
HDPE pipe a value for e can be expected to be similar to that of
ommercial steel pipe. Appendix B-1 indicates that for such a pipe,

= .00015 and thus, for a 42 inch pipe $\frac{e}{D} < .0004$. Appendix B-2
en shows that for a Reynolds number of 9.252×10^5, f = .0167. Head
ss due to friction in the pipe can then be calculated by Equation
0). Appendix A-1 shows the actual calculations.

PIPE WALL THICKNESS

The thickness of the pipe is crucial for lateral strength and iffness since it must resist the wave and current forces acting on it. The pipe must also resist pressure forces during the controld submergence during deployment and from the negative pressure from e seawater pump(s), which could collapse the pipe if it is not rong enough. The temperature gain in the cold seawater flow is also function of the pipe thickness, since the only insulation of the pe is expected to be from the pipe material itself.

With the pipe material determined, the required wall thickness be calculated from the forces acting on the pipe due to water moon, and from the pressure forces acting on the pipe wall. The pipe ckness must be great enough to prevent hydrostatic collapse. To demine the critical buckling pressure for the pipe, the following mula can be employed:

$$P = \frac{3EI}{(1-\mu^2)\ R_m^3} \cdot C \qquad \text{Eq. (11)}$$

where:

P = Design pressure

E = Modulus of elasticity

μ = Poissons ratio

I = Moment of inertia

R_m= Mean radius

C = Reduction factor for out of roundness

(1.0 in this case)

In this calculation, the modulus of elasticity must factor in creep modulus, because plastic material creep under long term con-nt load. A common value to use here is E_{50} = 200/.300 N/mm^2 hrock 1981).

The moment of inertia in this case is calculated by the formula:

$$I = \frac{t^2}{12} \qquad \text{Eq. (12)}$$

where t = pipe wall thickness

The calculation for the maximum stress and the maximum allowable ess is shown in Appendix A-2.

HDPE pipe is readily available in diameters up to 63 inches in foot sections. A pipe size of 1200 mm is selected for the seawater ply pipe. The manufacture's literature indicates that this size e is available in two wall thicknesses, indicated by SDR on the a sheet. SDR is the ratio of outside diameter to wall thickness. s shown by Driscopipe, a division of Phillips Petroleum, are 26 32.5; however, according to the sales representatives, other s are also available upon request.

The pipes used at Keahole Point vary in thickness. SDR's of 21 d 17 were used in areas where pressures were critical, or where re- stance to abrasion on the sea floor was required. According to one les representative, the pipe can be manufactured at any thickness sired.

For the purpose of this design, an SDR of 21 is selected for the tire pipe length. The calculations in Appendix A-2 use an SDR of 21 d width of 1200 mm pipe size, which results in a wall thickness of 7 cm. The negative pressure required of the seawater pumps to over- me the major head loss due to friction is calculated to be 1.06 x ^{5} N/m^2 (14.6 psi). This does not exceed the critical collapse pres- re of the pipe, which is calculated to be 1.83 x 10^5 N/m^2 5.5 psi). Therefore, the thickness ratio is acceptable. Two other icknesses were also examined. The results of these tests are as fol- ws in Table (3).

Table (3) Comparison of Pipe Sizes and Thicknesses

Pipe OD m (in)	SDR (OD/t)	Wall Thickness m	δP_{crit} N/m^2 (psi)	ΔP N/m^2 (psi)
1.200 (47.2)	21	.057	1.83 x 10^5 (26.5)	1.006 x 10^5 (14.6)
1.067 (42.0)	21	.051	1.83 x 10^5 (26.5)	1.809 x 10^5 (26.2)
1.200 (47.2)	26	.046	9.39 x 10^4 (13.6)	9.098 x 10^4 (13.2)

The second and third pipe thicknesses examined had very narrow gins of safety. Although, these may be adequate, since the modulus

based on a 50 year life expectancy already, other stresses from nding and twisting of the pipe have yet to be considered. Therere, the more conservative pipe design of 1.200 m OD and SDR 21 is lected.

The bending stresses on the pipe due to currents and wave motion ould also be considered. Experience on other pipeline projects ows that these forces can be expected to deflect the pipe between e anchored sections and also during deployment. The maximum allowle deflection is given by the equations below.

$$\delta_{max} = \frac{5w_oL^4}{384EI} \text{ inches} \qquad \text{Eq. (13)}$$

I is given by:

$$I = \frac{M}{Z}(R_o^2 + R_i^2) \qquad \text{Eq. (14)}$$

where:

I = Moment of inertia (in.^{4})

w_o = Force per unit length (kips/inch)

E = Modulus of elasticity (kpsi)

R_o = Outside radius of pipe (inches)

R_i = Inside radius of pipe (inches)

M = Mass of the pipe section, $M = \rho\ \pi\ (R_o^2 - R_i^2)$ (lbs)

The actual calculation is left to a more detailed analysis and ign.

TEMPERATURE GAINS

Calculating the temperature gain in the seawater flow could be simple as determining the overall heat transfer coefficient for pe in seawater and using it in equation (15) to determine the heat nsfer, then equating it with equation (16) to solve for the change temperature,

$$q = U_o A_o [T_o - T_1] \qquad \text{Eq. (15)}$$

$$q = MC_p \Delta T \qquad \text{Eq. (16)}$$

However, the temperature of the seawater outside the pipe is not nstant over the length of the pipe. This complicates the calcula-on, but experience from previous studies helps a great deal in the ea of temperature gain.

In the Miami Beach study (Hirshman and Kirklin 1977), the ex-ted cooling water temperature change calculation was based on a imum flow rate of 50,000 gpm (3153 lps), a significantly higher w than this Waikiki design. However, similar temperature and depth ditions to Waikiki make a comparison of methods reasonable. The water intake system was divided into several sections. External am-nt thermal conditions for each section were determined from avail-e data. Calculations for each section were completed using the ses. Temperature gains in each segment of the system were added cu-atively to determine the delivered temperature. Calculations were eated for different flow rates, and a minimum acceptable flow was termined.

The calculation was then initiated at the 525 m depth level, us-
g an inlet water temperature of 6°C as the fluid temperature out-
de the pipe for the first segment (T_o) and the first segment length
L). This segmental thermal conduction temperature change was de-
rmined and added to the original inlet temperature. This new inlet
mperature was used in the succeeding pipe segment temperature
ange calculation. Each succeeding incremental segment temperature
ange was obtained in this manner. The total temperature change due
conduction in the unburied section was then determined.

Almost the same method is used in the Waikiki case. To calculate
e change in temperature the pipe length was divided into several
ctions. These sections were chosen from the bathymetry and tempera-
re information and assigned average seawater temperatures outside
e pipe. The temperature increase was calculated over the length of
e first section of pipe and was added to the initial temperature of
e next section of pipe for the calculation of that section, and so
. Six sections were chosen. A sample calculation is shown in Appen-
x A-3. No heat gain due to friction was considered in this case.
ice from the experiences in other designs, it has not been a large
tor.

Three assumptions were made to simplify this calculation. The
st is that the force flow over the outside of the pipe from wave
current induced water motion is a constant 1 m/s (2 knots) in a
ection perpendicular to the pipe in the heat transfer calcula-

ons. Actual measurements of steady current in the intended area of
e project indicate a steady current of 1 knot is normal. A current
2 knots is assumed to add a margin of safety to the calculations.
more refined calculations, a more accurate analysis of water mo-
on over specific sections of pipe could be determined. This can be
nsidered a "worst case" since we know also that the motion will
obably not be perpendicular to the pipe everywhere. The second as-
mption is that C_p for seawater is constant over the length of the
pe, both inside and outside, even though from Appendix B-3 it is
own to vary with temperature. A third assumption is that the pipe
ll be exposed to the surrounding seawater everywhere on its route.
is will not be the case; some sections, such as in the surf zone
d in the canal and harbor, will probably require burial. The assump-
on that convection due to flow of water around the entire circumfer-
ce of the pipe is again a worst case assumption from a heat
nsfer standpoint for pipe laying on the bottom.

CALCULATION OF COLD WATER PUMPING POWER REQUIREMENT

The power requirement to pump the cold seawater can be calcu-
ed as follows:

$$\dot{W} = \rho \, h_l \cdot V \cdot A \qquad \text{Eq. (17)}$$

where:

$\dot{W}$ = power required

ρ = seawater density

h_l = head loss (note $h_l \cdot \rho = \Delta P$)

V = flow velocity

A = cross sectional area of pipe (inside) = $\frac{\pi(ID^2)}{4}$

As calculated in both Appendices A-1 and A-2, the velocity is
own to be 1.329 m/sec, ΔP is 6.022×10^4 N/m^2, and ID is 1.086 m,
e power required is calculated to be 74,113 W. If a pump efficiency
η = 80% is assumed,

$$\frac{74,113W}{\eta} = 92.64 \text{ kW}$$

1 hp = 746 kW

Therefore the pump must be at least 92.64 kW or 124.2 hp to de-
ver the seawater to the site at sea level.

The head loss and pressure drop considered here are only for the
pe losses. For each meter above sea level the water must be deliv-
ed, a simple calculation yields an additional 1,581 watts or 2.12
that will be required.

EFFECTS OF WAVES AND CURRENTS

The principal natural forces affecting a marine pipeline after
tallation are direct wave and current forces near the bottom.
se can have secondary effects when, for example, the sediment un-
a pipeline is eroded and the pipeline becomes non-uniformly sup-
ted. The pipeline is considered stable if the forces holding it in
ce are greater than the forces that would tend to move the pipe.
an unanchored, unburied pipeline resting on a level, stable

ttom, the forces holding it in place are only gravity and friction.

Wave forces at the bottom are due to both a steady current and oscillatory motion produced by the waves. This motion reverses itlf at half the wave period perpendicular to the wave front. In addion to regularly reversing drag and lift forces due to the rizontal water particle velocity, there is a horizontal inertial rce due to the water particle periodic accelerations. The inertial rces are opposite in phase to the drag and lift forces. Each of ese forces may be calculated separately, and whichever represents e larger force, since they are out of phase, at a particular depth used as the maximum design force.

The forces experienced by a submarine pipe will vary depending the depth of the water, currents, wave conditions, pipe size, ape, proximity to the bottom and to the free surface. The deep er sections of the pipe will see forces resulting from steady curts. The shallower sections will see forces resulting from wave acn, as well as currents. Because of the differing conditions over length of the pipe, several different systems of anchoring or supting the pipeline may need to be employed to ensure stability rshman and Kirklin 1977).

The stability and structural integrity of the submerged pipeline ends, to a great extent, on its stability after installation. In pipeline industry, the term stability refers to the ability of

e pipeline to remain as placed. If the pipeline moves or its sur-
undings are altered, its structural integrity is threatened. Much
fort is expended in the pipeline design to predict these forces and
en to design the pipeline for stability under the anticipated condi-
ons.

The degree of stability required is a function of the pipe mate-
al used and its properties, with regard to resisting deformation of
ilure in tension, buckling, external or internal pressure. Plastic
pelines can withstand an amount of movement and deformation that
uld destroy a steel pipeline; however, the weight in water of the
eel pipeline provides an inherent stability that is not available
th the lighter plastic pipeline. This is a decision the designer
st make early in the decision process. The decision has been made
the seawater air conditioning case to use HDPE.

It has been demonstrated by previous research (Hirshman and Kirk-
n 1977) that the forces caused by the design wave (hurricane condi-
ons) greatly exceed the maximum effect of the design current out to
oths of approximately 200 feet.

Considering winds, waves and currents, hurricane storm condi-
ons represent the most serious long-term hazard to the submarine
eline. A typical maximum deep water non-breaking wave of 50-foot
ght and 9 or 10-second period is commonly used as the design wave
rshman and Kirklin 1977). A maximum estimated bottom current of 2
ts (1 m/s) is commonly used in conjunction with the wave forces,

less site specific data indicates otherwise. As stated earlier, a eady current of 2 knots is probably much higher than would actually experienced by the pipeline, but such an assumption adds a large rgin of safety to the design. Under hurricane conditions, this may tually be low.

There are some basic techniques used to increase the stability the pipeline. These include increased weight of material, burial d/or other protective covering, and anchoring. It is common to bury e pipeline in shallow water where the forces and hazards, such as chors, are maximum, and to lay the pipeline directly on the bottom deeper water. Another technique, the catenary support, involves oring a free floating, buoyant pipe with a flexible cable which is chored to the bottom (Daniel 1989).

Reasonable assumptions for current and wave conditions for the rpose of calculations are a maximum bottom current of 2 knots run- ng parallel to the coast, the design wave of 15 meters and a 10-sec- l period in Appendix A-4. The calculations are made for a pipeline rpendicular to the coastline. The shelf and slope contours are as- ued to be parallel to the coast, and the wave front in deep water parallel to the pipeline.

In general, the forces acting on the pipe depend on both the e orientation, tides and wave conditions as stated previously. The ic hydrodynamics are derived from water particle kinematics. The ations for the particle motion in the horizontal plane are:

$$u_x = \frac{Hgk}{2\omega} \left[\frac{\cosh(k(d+z))}{\cosh(kd)}\right] \sin(\omega t) \qquad \text{Eq. (18)}$$

$$\dot{u}_x = -\frac{Hgk}{2} \left[\frac{\cosh(k(d+z))}{\cosh(kd)}\right] \cos(\omega t) \qquad \text{Eq. (19)}$$

where:

u_x = particle velocity in the horizontal direction

$\dot{u}_x$ = particle acceleration in the horizontal direction

H = wave height

g = gravity

k = wave number

ω = angular velocity of wave, $\frac{2\pi}{\text{wavelength}}$

d = depth

These forces may act both in the horizontal and vertical direc-ons. In addition to the hydrodynamic forces introduced by the mo-on of the water flowing around the outside of the pipe, the buoyant rce of the pipe itself must be taken into account. Since the pipe's nsity is 0.954 g/cc (Rubin 1990), the buoyant force is a factor.

)escription of Forces

Conventional engineering practice is to use the following equa-ons to calculate the hydrodynamic forces on a pipeline associated h wave and current action (Verley, Lambrakos and Reed 1989). For a g cylinder, or pipe of diameter D submerged in a fluid of density the horizontal force, F_x , and the vertical force, F_z , are given

$$F_x = C_D \frac{\rho D l}{2} u |u| + C_M \rho \frac{\pi D^2}{4} l A \qquad \text{Eq. (20)}$$

where

C_M = inertia coefficient

C_D = drag coefficient

u = undisturbed fluid velocity

A = fluid acceleration, $\dot{u}_x$ or $\frac{du_x}{dt}$

$$F_z = C_L \frac{\rho}{2} D l u^2 \qquad \text{Eq. (21)}$$

where

C_L = lift coefficient

Often the coefficients are assumed to be constant, but they are ctually time dependent.

The first equation (Eq. 20) is known as Morrison's equation. The drodynamic coefficients can be taken from various references as de-rmined from both potential theory and experimental data. The drag l lift equations utilize coefficients of drag, lift and inertia t are a function of pipe shape, Reynolds number, proximity to bot- and other factors. These are experimentally derived. The coeffi-nts that are used are from a cylindrical pipeline resting directly the bottom. Common values used in this situation are:

coefficient of drag, $C_D = 0.5$

coefficient of lift, $C_L = 1.4$

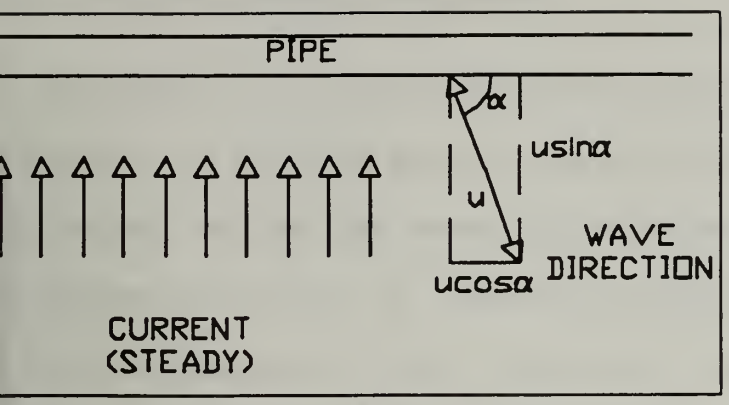
coefficient of inertia, $C_M = 3.3$

Experiments and field data have shown that these two equations edict the maximum hydrodynamic forces fairly well for the case of gular waves and a steady current. However, for the case of irregu- r waves and current, the variation between predicted and measured rces has been much greater. To describe the force time series for ch combined flows, other models have been developed. For the pur- se of this conceptual design, however, the basic model will be used.

It is important to understand how the different forces acting on the submerged pipeline interact. There is drag force in the lat- al direction, consisting of both frictional and inertial effects of ves, and friction from currents. These are represented by the first l second terms in Eq. (20), respectively. There can also be a lift rce on the pipe, as indicated by Eq. (21).

The orientation of the pipeline to the currents and direction of waves determines, to a large degree, what effect those forces e. If the pipe is oriented perpendicular to the flow of water over around it, those forces are greater. If it is oriented in a paral- direction to the currents and wave direction, the forces are atly reduced. See Figure (11).

There are several parameters which may be considered in determin- the hydrodynamic forces on the pipeline. Probably the most impor- one is the Keulegan-Carpenter number, $\frac{UT}{D}$, where U is the icle velocity, T is the wave period, and D is the pipe diameter.



g. 11 - Current and Wave Direction

If the Keulegan-Carpenter number is much less than 1, the inertial effects will dominate. This means that the water particle acceleration will be the major contributor to forces on the pipe. If the Keulegan-Carpenr number is much greater than 1, the frictional or viscous effects ll dominate, which means the water particle velocities will be the jor contributor to the forces on the pipe.

There is a dependence of forces, due to passing waves, on the raof pipe diameter to period and wavelength of incident waves. Genally, in the case where $\frac{H}{D} < 1$ and $\frac{D}{L} \geq 0.15$, viscous forces can be nored, and use of potential theory is recommended over equations) and (21). If $\frac{H}{D} \geq 1$ and $\frac{D}{L} \leq 0.15$, the empirically derived Morri's equation (Eq. 20) can be applied. This relation considers the cous effects. In cases where $\frac{H}{D} < 1$ and $\frac{D}{L} < 0.15$, other approaches be used. With pipelines and other slender members, such as piles, wave fields, the second case is relevant. This is why the industry ndard is to apply Morrison's equation (Eq. 20) for the lateral ces.

orce Analysis of Various Configurations

In addition to the dimensions of the pipe with respect to incom- waves, the way the pipe is installed or configured can have a eat effect on how the water flows around it, and hence, the resul- t forces acting on it. There are many ways to install a pipeline, for the purpose of this discussion, they can be broken down into e basic configurations as follows:

(I) The pipe may lie flat on the bottom in some sections

(II) The pipe may be partially buried

(III) The pipe may be completely buried

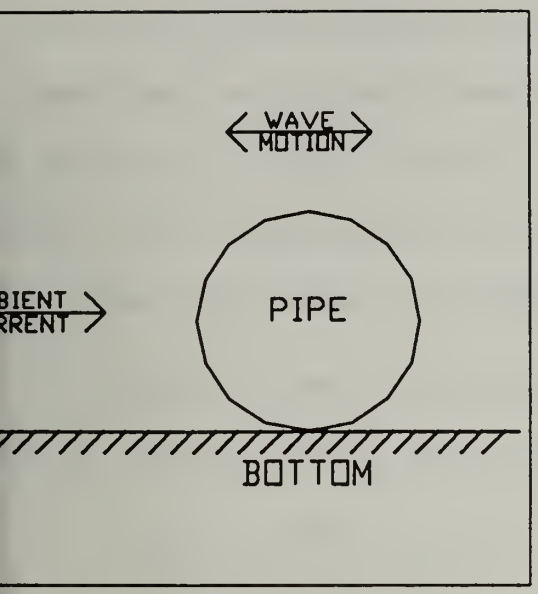
(IV) The pipe may lie on an uneven bottom with a small spaces under some sections

(V) The pipe may be suspended above the sea floor, either on legs or free floating and anchored by cables. This is representative of the catenary support system.

It may be useful to look at each one of these configurations in llow, intermediate and deep water. However, some of these can obvi- ly be eliminated from analysis.

ermination of Forces for Configuration (I):

In the configuration shown in Figure (12), the pipe lies flat on bottom. Although the design for the air conditioning system does include this configuration, analysis of this type of installation nstrates all the forces acting on each of the other basic configu-



g. 12 - Configuration I, Pipeline Flat on Bottom

rations. This is often considered the worst case for designers. At this point it is useful to look at a free body diagram to illustrate how all the forces act on the pipe. See Figure (13).

Drag Force:

In this case, the drag force is given by Morrison's equation (Eq. 20), and as shown on the freebody diagram (Fig. 13) as F_D. The flow over and ound the pipe can be broken down into two components: the component duced by the wave action, and the mponent induced by the steady cur- nt. As previously stated, particle ions due to wave motion in the hori- tal direction are given by equa- ns (18) and (19).

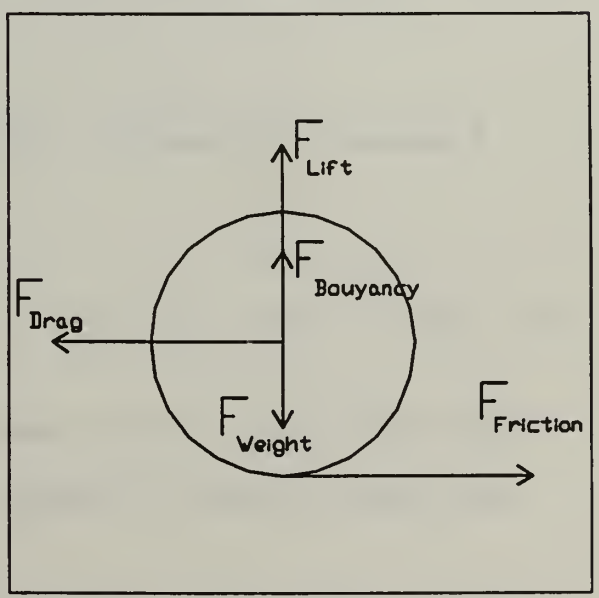


Fig. 13 - Free Body Diagram

The motions are time dependent 90° out of phase as the wave ses. It is important to see how the nitude of the viscous term in (20), the drag portion, is depend- on the square of the particle velocity given by Eq. (18). This is

contrast to the inertial term, the second term in Eq. (20), which dependent on the magnitude of the particle accelerations. Since e two terms in the lateral force are 90° out of phase, the maximum scous force and drag force will not occur at the same time. In ct, they will occur at opposite quarter wave periods.

A steady current alone also creates a viscous drag force, but th no inertial term. This force is given by:

$$F_D = C_D \frac{\rho D l}{2} u\,|u| \qquad \text{Eq. (22)}$$

As can be seen by the mathematical relationship between the drag rce and the square of the velocities, the drag forces cannot be ded directly to get the total force. Instead, the sum of the ve- cities should be substituted into Eq. (20). This equation then be- mes:

$$F_x = C_D \frac{\rho D l}{2} \cdot \left(u_{wave}(t) + u_{current}\right) \cdot \left|u_{wave}(t) + u_{current}\right|$$

$$+ \text{ inertial term}$$

$$= C_D \frac{\rho D l}{2} \cdot \left(u_r(t)\right) \cdot \left|u_r(t)\right| + \text{inertial term} \qquad \text{Eq. (23)}$$

This term is part of the overall F_{drag} shown on the free body gram (Fig. 13). The term l for pipe length is removed from the ht side of the equation.

t Force:

The F_{lift} shown on Figure (13) also depends on both the wave gen- ted particle velocities and the steady current. Again, as with the

ag forces, the wave incurred particle velocities are time depend-
t. Equation (21) is used, where $u = u_{current} + u_{wave}$.

oyant Force:

The buoyant force, $F_{buoyancy}$ in Fig. (13), is calculated by multi-
ying the volume of the pipe by the density of the seawater it dis-
aces. On a per unit length basis, only the cross sectional area is
ltiplied by the density as shown below:

$$F_B = \rho_{water}\, g \left(\frac{D_o}{2}\right)^2 \qquad \text{Eq. (24)}$$

ight Force:

The weight of the pipe, shown in Fig. (13) as F_{weight}, includes a
mponent from the mass of the water in the pipe and the mass of the
pe material itself, as shown below:

$$F_{weight} = \rho_{water}\, g\, \pi \left(\frac{D_i}{2}\right)^2 + \rho_{pipe}\, g\, \pi \left(\frac{D_o^2 - D_i^2}{4}\right) \qquad \text{Eq. (25)}$$

where:

D_i = inside diameter

D_o = outside diameter

ρ_{water} = seawater mass density

ρ_{pipe} = pipe material mass density

In the vertical direction, the forces consist of the buoyant
ce of the pipe and the opposing weight of the pipe, both of which
r constant, and the lift force, which is dependent on time. The re-
ut of these is a normal force on the seafloor. If the normal force

negative in sign, however, the pipe would of course be accelerated
ward the free surface. In that case, the normal force would have to
artificially increased in some manner for the pipe to remain sta-
e.

ctom Friction Force:

The resultant normal vertical force is a key factor in the total
rizontal force for the pipeline laying on the bottom, since the
iction force between the bottom of the pipe and the surface of the
afloor is dependent on the normal force. Therefore, this force is
so time dependent.

$$F_{friction} = fF_N \qquad \text{Eq. (26)}$$

where:

f = coefficient of friction

F_N = normal force

The coefficient of friction, f, differs for different bottom con-
ions and pipe intervals. This could be determined experimentally,
since bottom conditions can vary so much over the length of a
eline, it is difficult to know what coefficient to use unless all
conditions are known. Without specific data concerning the fric-
n characteristics of the bottom, a conservative value for the coef-
ient of friction of 0.5 is used in industry for marine sediments
rshman and Kirklin 1977). Thus, if the pipeline weighs 100 pounds
linear foot in water, it can be moved laterally on the bottom by
orce of greater than 50 pounds per foot.

The sum of the forces in the horizontal direction includes the rces calculated from equation (20) plus the friction force calcu-ted by equation (26).

To determine if the pipe is stable, all of the forces can be cal-lated at various time intervals over the wave cycle to determine e forces in the worst case, or maximum combined forces. If the pipe determined not to be stable, concrete or other types of weight may added to increase the downward force on the pipe, or some type of chor ing could be attempted to accomplish the same thing. Any weight ded would result in increased weight minus the buoyancy of that ma-rial. Appendix A-4 includes a sample calculation of the forces act-g on a pipe, configured as shown in Figure (12).

If the pipeline is anchored at regular intervals utilizing embed-nt-type anchors (into soil or rock), the computations are somewhat fferent than for stabilization solely by weight. In this case, the hors prevent the pipeline from sliding horizontally. The spacing the anchors for an unburied pipe is determined first by calculat-the maximum vertical lift component of wave and current forces ex-ted at a particular depth. The vertical force required for l-out of each anchor set is determined, and a safety factor is ap-ed. A maximum spacing calculation to prevent pull-out can also be termined. A further calculation is performed, using the maximum vec-sum of wave and current forces, to determine, for a particular h and anchor spacing, if the deflection of the pipe between an-rs due to these forces can cause failure or damage to the pipe.

s is a function of the pipe material and design. The limiting safe stance between anchors can be either pull-out or deflection-limed, depending on circumstances at a particular depth.

ermination of Forces for Configuration (II):

A similar analysis of forces can be done on the pipe in Figure). The equations used in the first case can be used again with y slight modifications. The forces in the z-direction would change tle. Lift force would be estimated in the same manner as would ght and buoyancy. Although the situon is not exactly the same, the e coefficient of lift could be used obtain a reasonably good estimate the lift forces.

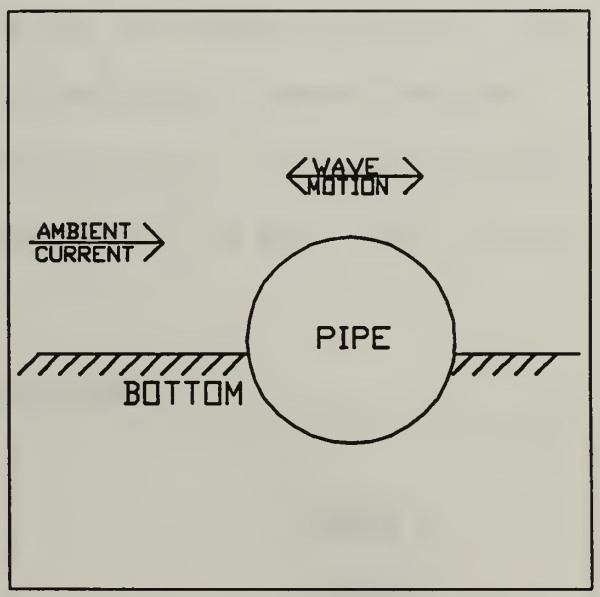


Fig. 14 - Configuration II, Pipeline Partially Buried

The inertia and drag forces in x-directions, determined by equan (20), would be significantly s. The viscous drag term could be uced by a factor of 2 since only ½ the diameter D is exposed to the icle motions. The reduction in diameter has an even greater ef- on the inertial term since, in that case, the diameter is ired. It is immediately obvious that the frictional force may be ter, depending on how deeply the pipe may be buried.

Partial burial is not anticipated to occur in the design of the vention center air conditioning system; however it is a common con- uration of submarine pipelines. This is often different than the ginal configurations of the pipeline, since the action of the es and current may redistribute the seabed around the pipe. The ces required for break-out of the pipe in the semi-buried state de- d on soil characteristics and permeability, as well as the charac- istics of the hydrodynamic forces. The break-out of the pipeline ies on a couple of mechanisms. If the wave induced lift force and yancy force exceed the weight of the pipe, an upward force will re- t. At this point, there are two forces resisting upward response the pipe; these are the shear stress between the pipe and the sand sediment grains, and the negative pressure of the pore water below pipe (Foda, Chang and Law 1990).

ermination of Forces for Configuration (III):

In case III, the pipe is com- tely buried (see Fig. (15)). The de- n of the air conditioning cold er supply pipe is intended to in- le burial of the pipe in the canal, harbor and through the surf zone. is considered the best way to d damage to the pipeline. The hy- ynamic forces are less relevant e they are reduced to only the

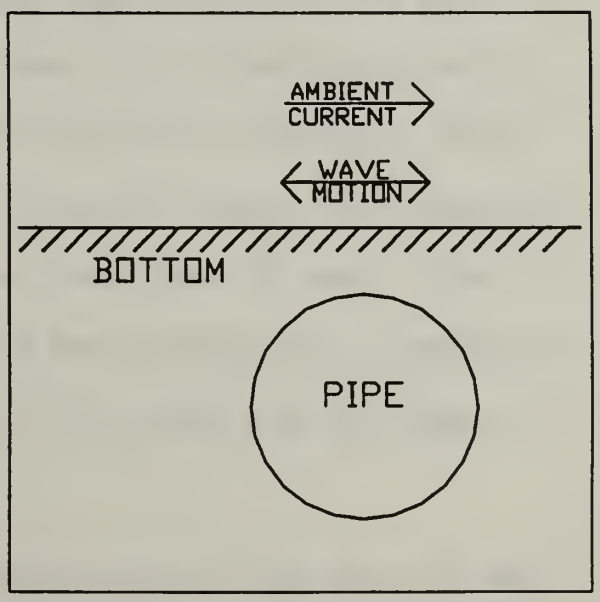
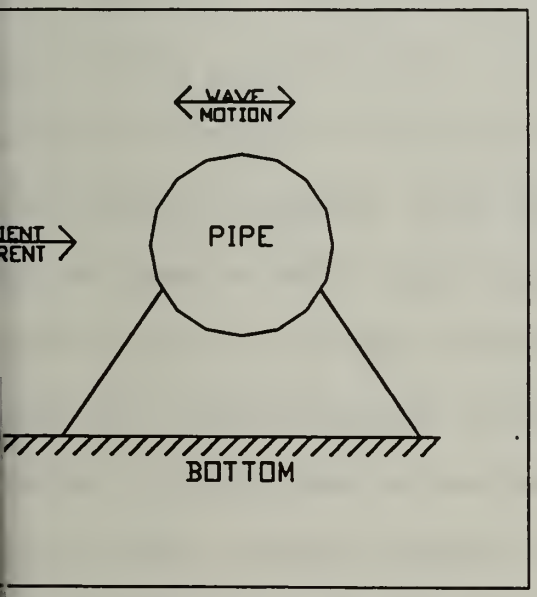


Fig. 15 - Configuration III, Pipeline Completely Buried

yancy and the weight of the pipe; it must simply be heavy enough to float in the sediment. The forces here also may vary depending the qualities of the bottom. The bottom sediment could liquefy un- some wave conditions. This case requires the study of seafloor l mechanics.

ermination of Forces for Configuration (IV):

In case IV, the pipe is suspended on top of some sort of support ucture (see Fig. 16). This is the intended method of pipe installation off Waikiki beyond the surf zone out to a depth of approximately 60 meters. If the pipe can be supported one diameter or more above the sea floor, the lift force can be neglected. The reason for this is if the particle motion around the bottom of the pipe is nearly identical to the particle motion over the top of the pipe, then the two forces are equal and opposite, with a net resultant of zero. This is neglecting any difference in particle ocities between the depth at the top of the pipe and the depth of bottom.



. 16 - Configuration IV, eline Supported Above floor

The supports would probably be concrete, as in the case at Kea- Point. They serve duel purposes by adding weight and reducing

t. In this case the normal force resulting from the difference be-
en combined buoyancy and the combined weight of the pipe and sup-
t is the total normal force used to determine the friction force,
ch acts to resist the drag and inertial forces in the lateral di-
tion (x-direction).

ermination of Forces for Configuration (V):

In the case V, as in case IV, lift forces on the top are negated the identical force acting on the tom of the pipe (see Fig. 17). s type of support is representative the catenary support used success-ly at Keahole Point. This type of port has all the same advantages case IV. One additional force to sider in this case is the forces ing on the flexible support. These be calculated in a similar fash-to the forces on the pipe.

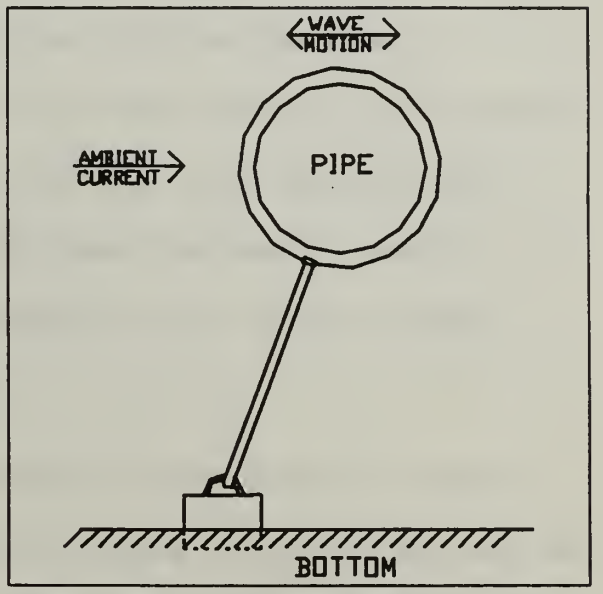


Fig. 17 - Configuration V, Catenary Mooring of Pipeline

Again as in the fourth case (Fig. 16), the weight or anchor acts overcome the excess buoyancy force in the z-direction and the drag any inertial forces in the x-direction. This type of support d probably only be used in deep water, therefore the forces would result from the steady current present. With no particle veloci-

s and accelerations from wave action, the steady velocity is only
m the current, and the inertial term is neglected.

Picking coefficients such as the drag coefficient C_D and added
s coefficient for the forces in the x-direction can also be diffi-
t. Just as in the case of lift, the forces affecting the pipeline
dependent on distance from the bottom, depth, marine growth, cur-
t conditions and wave conditions. Different values for the con-
nt could be determined experimentally for all the various
ditions, and researchers have tried to do this. Some of the experi-
ts done in the area focus on finding the RMS (root mean square)
ues of those coefficients for given Keulegan-Carpenter numbers. In
ther approach, the forces can be decomposed into fourier compo-
ts (Shankor, Cheong and Subbiah 1988).

There is a great deal of data published on the subject, with a
ber of different approaches to determine these coefficients and im-
ving the predictions for Morrison's equation and variations of the
ation. Many contradict the predictions of the Morrison's equation,
most of these are not as easy to use.

cour

A pipeline laying on the seabed may lie flat or there could eas-
be areas under the pipeline where flow could pass due to the un-
seafloor. These areas could be subject to erosion of the seabed
to the water motion and scour. Scour under and around the pipe-
can alter the seabed upon which the pipeline relies for stabil-

y. Prediction of scour below and around pipelines is a major focus pipeline engineering practice. This is an important area to con-der when a pipeline is to be exposed to currents or waves, and po-ntial theory is unable to explain or predict scour depth (Sumer, nsen and Fredsøe 1988).

In a steady flow, the scour can be expected under and on the lee-rd side of the pipe, but in oscillatory flow the scour of one half riod is the upstream part of the next half period. The scour depth pends on several factors. The size and specific gravity of the sedi-nt particles which comprise the seabed play a role, as does the con-lidation of the bed, just as in the case of longshore transport.

Although water particle motions from a steady current and wave tion passing beneath the pe may exceed the critical ocity required for the on- of sediment transport in vicinity of the pipe, s is not required for ur to occur. The focus of e studies of scour around elines is the vortex shed-g on the leeward side of pipe (see Fig. 18).

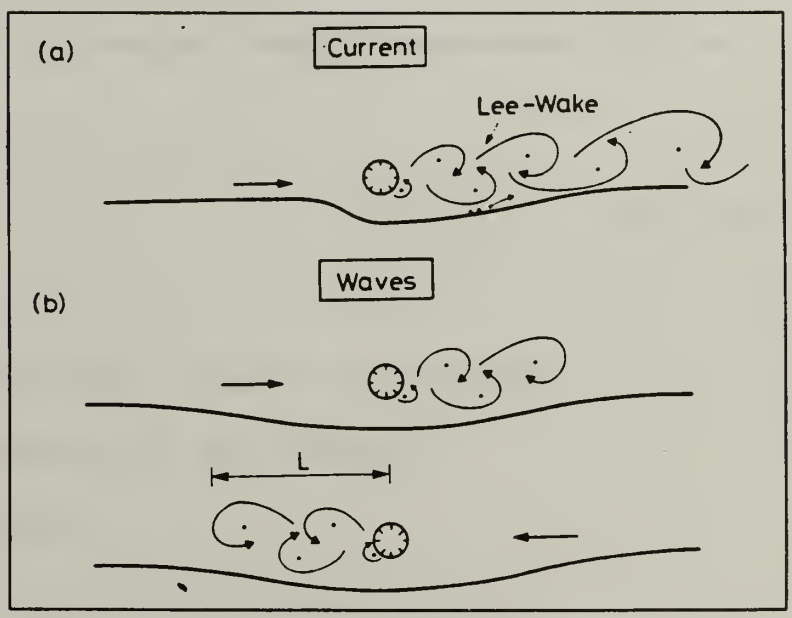


Fig. 18 - Lee-Wake Effect: (a) Currents, (b) Waves

As the flow exceeds the critical Reynolds number and flow around e pipe becomes turbulent, vortices form on the leeward side of the pe. As the flow velocity increases, the vortices begin to shed into e wake in an oscillatory manner. The velocity of the particles in e vortices can then exceed the critical velocity required for the set of sediment transport.

The vortices can be characterized by St, the Strouhal number:

$$St = fD/U \qquad \text{Eq. (27)}$$

where:

U = the mean flow velocity

f = the vortex shedding frequency

D = pipe diameter

The shields parameter θ_o is another useful measurement of the ow around the pipe:

$$\theta_o = \frac{U_f^2}{g\,(s-1)\,d} \qquad \text{Eq. (28)}$$

where:

U_f = undisturbed shear velocity on the bed

s = specific gravity of the sediment

d = sediment size

g = gravity

The depth and position of the scour depend on the Strouhal num-, shield parameter, height of the pipe from the seafloor and soil racteristics.

The Keulegan-Carpenter
mber is considered the gov-
ning parameter of the scour
ench depth (Sumer and Fred-
e 1990 and Chiew 1993).
ves and currents affect
our differently as Figure
9) illustrates.

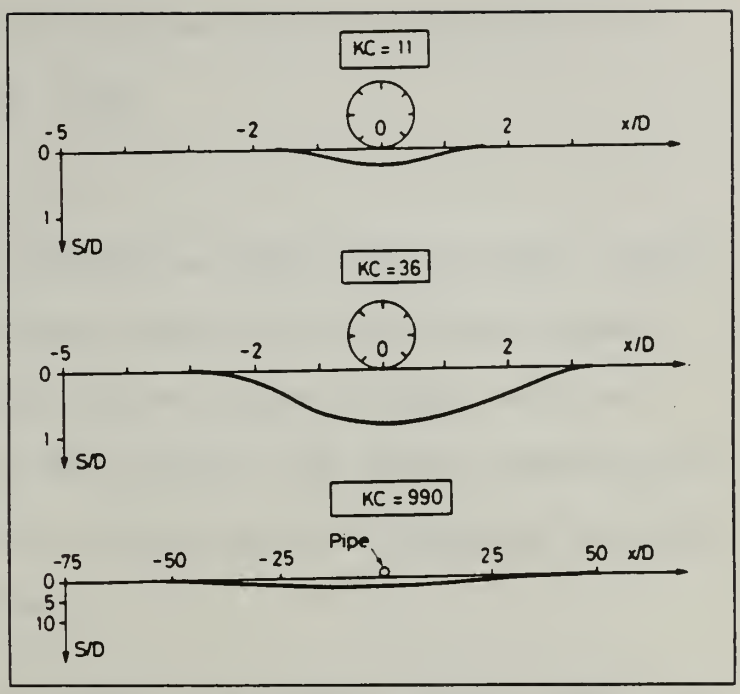


Fig. 19 - Equilibrium Scour Profiles for KC = 11, KC = 36 and KC = 990

Sumer and Fredsøe (1990)
esent an relation for a pipe
control with a bed,

$$\frac{S}{D} = 0.1\sqrt{KC} \qquad \text{Eq. (29)}$$

where:

S = final scour depth below the pipe (max)

D = diameter of pipe

KC = Keulegan-Carpenter number = $\frac{UT}{D}$

The trench reaches an equilibrium depth when it is large enough
t the flow velocities are decreased to below the critical veloci-
s for initial scour (θ_{crit}). As the Keulegan-Carpenter number in-
ases, the scour increases both in depth and length. Roughness of
pipe has almost no effect. The scour beneath the pipe is known as

e tunnel scour. The scour occurring downstream of the pipeline is own as the lee-wake scour (Chiew 1993).

Vibrations

Not only can the bed around a pipeline erode due to scour, vibrations of the pipe are induced by both wave action and vortex shedding. The wave action produces lift forces which, as seen earlier, cillate in amplitude during the wave period. The vortex shedding inces changes in pressure acting on the pipe with a frequency at that the oscillations of the vortices.

There may also significant changes in the force coefficients, C_D d C_M due to the vibrations induced on a pipe. The drag coefficients nd to be larger for vibrating cylinders, partly due to an apparent crease in diameter and partly due to an increase in the velocity. e inertia coefficient, C_M, decreases relative to a fixed pipe (not brating) attributable to the pipe hitting its own wake. The effect the coefficient of lift does not appear to be significant (Sumer, dsøe, Gravesen and Bruschi 1989).

Discussion of Combined Effects/Pipe Design

The industry standard for predicting forces in submarines pipees is the use of Morrison's equation (Eq. (20)) and Eq. (21) for t forces. However, there are serious doubts as to the accuracy of h an approach when used alone. The use of equations (20) and (21) also made more difficult in some pipe configurations, such as par-

ally buried pipelines; they are not applicable at all to buried pelines.

The effect of scour and vibration should also be taken into account when designing a pipeline. Not only should the forces and responses for intended installation configurations be considered, but e forces and responses associated with any anticipated changes in e configuration should be taken into account as well. Tunnel scour d lee-wake scour contribute to changes in the configuration of the pe and vibration due to vortex shedding could add to the effect.

Empirical and analytical analysis of a design should be supported with model testing, if possible, before going forward with a sign. Taking all the force considerations into account, the pipeline design should include three of the configurations mentioned.

The nearshore sections of the cold water supply pipe and effluent pipe located inside the reef, in the Ala Wai Boat Harbor and Canal, will all be buried. This would require a significant dredging effort. Since the channel is already dredged for navigation, the pipe the boat channel would have to be buried to a depth at which future maintenance dredging for the the channel would not damage the pe. Along with burial, some form of securing the pipe to the bottom the trench is advised. This could be accomplished be bolting the e down with clamps if there is sufficient basaltic rock or coral ow into which bolts could be driven. Another method could be to ght the pipe down with concrete. This section of pipeline would be

a total length of 2500 meters, extending out to a depth of approximately 20 meters.

From the 20 meter depth to approximately a 100 meter depth, the pipeline could be anchored by concrete blocks attached to the pipe. This method was used successfully at this depth at Keahole Point. The blocks and anchors would support the pipe at least one diameter above the sea floor to minimize the hydrodynamic lift. A calculation similar to that in Appendix A-4 could determine the amount of concrete per unit length of pipe which would be required for the pipe to remain stable for this section. This section would extend approximately 900 to 950 meters offshore, beyond the buried section.

The remaining section of cold water supply pipe could use the unique buoyant catenary support, as designed by Makai Ocean Engineering Inc., Waimanalo, Hawaii to avoid the rough bathymetry of the steeper sections. This was used at depths of 160 meters and greater at Keahole Point, and has proven successful. Off Waikiki however, the sharpest drop off begins at about 100 meters. Considering bottom contour, steepness, and roughness of the bathymetry, this is the desired depth at which the use of catenary supports should initiated.

This could be a hazardous area in which to make a transition to the more fragile catenary support, depending on the wave conditions encountered at this section. At this distance offshore, waves may encounter the pipe section at adverse angles (close to perpendicular). Calculating the wave forces on the pipeline by the

method in appendix A-4 however, indicates that the lateral force acting on the pipe due to the incident design wave is negligible. This is to be expected since the design wave of height 15 meters and period of 9 seconds would have a wavelength of 126 meters and in 100 meters of water, these are deep water waves.

The details of the bathymetry are important to the decision where to make the change to the catenary support. A small change in the route of the pipeline could change the depth at which the catenary can be safely employed. For these reasons, a particularly thorough investigation of the bathymetry and bottom conditions is recommended along this section of the pipe route prior to final design.

NVIRONMENTAL CONSIDERATIONS

The seawater cooling concept calls for a single pass of the cold water through the seawater-chilled water heat exchanger, raising temperature of the seawater some 5^0C. Once through the heat exchanger, the seawater is essentially high quality waste water, still cooler than ambient surface water. In the design of the system, -toxic and mostly non-metallic materials are utilized. The discharge water is clean seawater, with excellent water quality characteristics.

Discharge outfalls from the convention center to the ocean would duce no net positive or negative environmental impact because of small discharge volume compared to the size of the receiver body the good mixing in the nearshore region. Using a diffuser at the charge end of the outfall would bring the water to within a few de- es of ambient at the discharge point, and since the quality of the charge water is excellent, no adverse effect is anticipated. The an outfalls, however, will be expensive because they will have to designed to withstand the rigors of the surf zone and will fall o the expensive offshore construction catagory.

OULING

In the ocean intake pipeline, fouling in the offshore pipeline not expected to significantly reduce flow efficiency. The cold water is nutrient rich, however, research at Keahole Point, Ha- i, has shown that bio-fouling will only occur if the water is ex- ed to sunlight. Exposure of the water to sunlight is avoided in s design. Severe internal fouling is not likely to occur unless system is shut down for a long time, in which case fresh water or orinated water could be sealed in the pipeline. Often, just seal- the pipe causes the system to turn anaerobic, killing the aerobic anisms. The pipline could permit cleaning "pigs" of the type used oil pipelines or other devices, to be pulled through. However, is is not expected to ever be necessary.

ALTERNATE DESIGN: DIRECTIONAL DRILLING

An alternate technology which could be employed in lieu of dredg- the nearshore sections of pipeline is directional drilling. This loys machine drilling equipment to drill shafts which may run hori- tally or at any desired angle for as much as a mile. The technol- currently allows the drilling machine to be guided in a manner ch that it can curve upward and outward and break through the sur- ce at any desired target point.

Directional drilling has been in use in the gas and oil pipeline dustry for twenty years, and more recently used by the telecommuni- ions industry.

Directional drilling may be a desirable method whenever the envi- mental or economic impacts of open drilling are unacceptable. fts can be drilled under highways, industrial areas, rivers or any face obstacle which cannot be disturbed.

The equipment used is similar to standard drilling equipment and loys sensors in the drill bit to measure compass heading and an- s of tilt and rotation. The computer guided drill head is accurate within 1% of the shaft length (Wilkins, Muller 1991).

The problem which first comes to mind is the inhomogeneities ch exist in the volcanic terrain, which makes up the Hawaiian geol- . However, this and many other technical problems were overcome in emonstration of directional drilling at Keahole Point, Hawaii

February, 1992. At that time, a shaft of 3.75 inch diameter was ccessfully drilled a distance of 610 feet through inhomogeneous vol- nic rock to a point 320 feet offshore, at a depth of 30 feet, where broke out. The test shaft was then reamed out to a diameter of 26 ches, for a distance of 20 feet when the demonstration was consid- ed a success. It is believed that shafts could be reamed out to a amter of as much as 54 inches with the present technology.

The drilling technology could conceivably be used in the near- ore surf zone for both the cold water supply pipe and the effluent pe. The shafts could still be lined with the 1200 mm diameter high nsity polyethylene pipe. This would necessitate some type of joint tween the offshore section of the pipe and the drilled section.

The advantages would be the superior pipe protection from cur- nt and wave forces on the pipeline in the surf zone. It might also eviate many of the evironmental problems associated with dredging rench for the pipeline. Any dredging activity in that area would subject to intense scrutiny from environmentalists and the visitor ustry in Waikiki.

However, there are disadvantages as well. The drilled shaft ld probably have to begin at the convention center site itself, to space constraints in the area. This would necessitate a lled shaft length of approximately 2500 meters or more. This is n longer than has been proven technically feasible to date.

t may not be possible. Another disadvantage is the expense. The st to drill a tunnel of that length and diameter would be much eater than the more conventional alternative.

Although the drilling technology is an interesting option, at is point, there are too many unknowns to propose it as a viable thod of pipeline installation in this application. For these rea- ns, the more conventional methods of installing the pipeline are in- nded for use on this project.

SEAWATER HEAT EXCHANGERS

Heat exchangers are available in a variety of types, sizes and erials. There are standard models available by many manufactures, vever, a single unit of the size required for an air conditioning lication of this size would probably be custom made.

One available design is called a finned tube heat exchanger. The water could pass through tubes that have fins attached over which fresh water would pass. The tubes or coils could be constructed titanium, thus resisting corrosion by the seawater, and the fins ld be aluminum.

Another design considered is a plate and frame heat exchanger. se are made of an assembly of pressed metal plates, aligned on or ured to metal frames or bars, and a containment or cover. Gaskets set in the outside groups to contain the fluids and to direct the id flow distribution. The hot and cold fluids can flow in the oves in the plate in opposite sides of the plates. This is the e selected in this air conditioning design.

There are many advantages to plate and frame heat exchangers. principle advantage is the accessibility of the heat exchange sur- es for cleaning. Other advantages claimed by plate and frame heat changers over tube and shell exchangers are as follows:

1. Less surface area required for heat transfer, resulting in cost savings. The close spacing of the

grooves between adjacent plates result in a geometry similar to that of very small tubes.

2. Less space required for the unit(s) as a result of the reduced surface area.
3. Better heat exchanger effectiveness.
4. The number of plates can be easily increased or decreased depending on future load changes.

The only disadvantage over other types is the large number of faces sealed by gaskets. Due to the use of elastomeric materials the gaskets, the internal temperature and pressures of the heat exnger are generally limited to 300 °F and 400 psi; however, both well within the range of this air conditioning application.

laterials

In a series of experiments and corrosion tests at NELH, ALCAN International has determined that the heat exchangers for deep cold water application can be constructed of aluminum alloy. The water ting corrosion, which is normally the result when exposed to 6°C p seawater, can be controlled by introducing a thin surface layer zinc to the aluminum. This limits the pitting to the zinc layer. t exchangers constructed of this zinc clad aluminum alloy are less ensive than the titanium units and are expected to last 10 to 20 rs.

Bio-fouling

One of the major concerns with seawater heat exchangers has been o-fouling. However, as with the cold water supply pipe, research s shown that bio-fouling on the cold water side of the heat exchang- s is virtually nonexistent as long as the cold, nutrient rich seawa- r is not exposed to sunlight.

Design

The type of heat exchanger, the material used, the heat transfer quired, and the temperature differences of the fluids on both the ot" and "cold" sides determine the size and cost of the unit or its.

The size is determined by the surface area required over which e fluids must flow. The amount of thermal energy transferred is de- rmined by:

$$q = K \cdot A \cdot \Delta T_m \qquad \text{Eq. (30)}$$

where:

q = the desired amount of thermal energy transferred

A = the heat transfer area

K = the overall heat transfer coefficient

ΔT_m = the log mean temperature difference (LMTD)

The desired thermal energy transferred is the result of the ange in temperature desired in the fluids multiplied by their spe-

ic heats and the mass flow, $\dot{m}$,

$$q = \dot{m} \cdot C_p \cdot (T_{in} - T_{out}) \qquad \text{Eq. (31)}$$

The effective area of the heat transfer is a function of the ge-
try of the heat exchanger. It represents the surface area exposed
the fluid.

The overall heat transfer
efficient, K, is also a func-
on of the geometry, and the
terials and fluids used.
s is a characteristic spe-
ic to the application, and
often determined by pre-

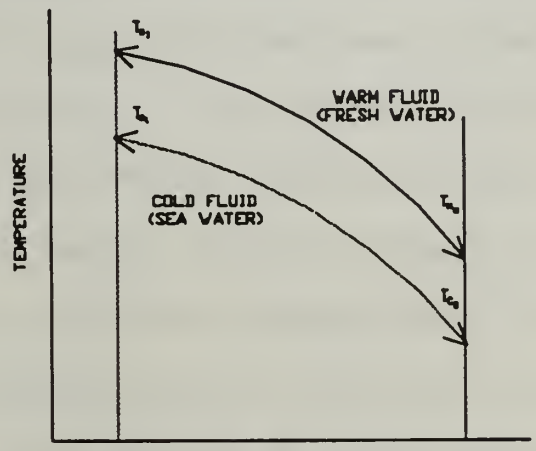


Fig. 20 - Log Mean Temperature Difference

us experience. This is adjusted down for a fouling factor which ac-
nts for the reduction in effectiveness over time due to fouling of
grooves or tubes in the heat exchanger.

The log mean temperature difference is the temperature parameter
which heat exchanger performance is based. It is defined by the
perature difference at the other end of the heat exchanger divided
the natural logarithm of the ratio between the two temperatures.
temperatures are illustrated in Figure (20) for a counterflow
exchanger.

LMTD is calculated as follows:

$$LMTD = \frac{(T_{H_2} - T_{C_2}) - (T_{H_1} - T_{C_1})}{\ln \left[\frac{(T_{H_2} - T_{C_2})}{(T_{H_1} - T_{C_1})} \right]} \qquad \text{Eq. (32)}$$

In the case of the seawater air conditioning system, the LMTD is all, about 0.95 °C. Therefore, a high heat transfer coefficient is sirable to reduce the area required as much as possible.

Zinc-clad aluminum plate heat exchangers of the type required uld probably have to be custom built and custom designed. Titanium at exchangers are available "off the shelf" from companies such as fa-Laval Marine & Power and Mueller Accu-Therm. Aluminum alloy heat changers are also available "off the shelf." ALCAN of Canada is one nufacturer which has had success with aluminum seawater heat exangers at Keahole Point, Hawaii. Aluminum has the advantage of be- much less expensive than titanium, however, it does not last as ng. Life expectancy for aluminum alloy heat exchangers might be out 20 years, as opposed to 30 or 40 years for titanium. It can be pected that aluminum heat exchangers and titanium heat exchangers uld be in competition for a job of this size, and economics would ide which type and models would be chosen, provided they could adetely meet the performance requirements. Table (4) summarizes the t exchanger requirements. Such data can be supplied to vendors and ufacturers who could respond with options for "off the shelf" dens in the case of titanium heat exchangers.

Table (4) Heat Exchanger Design Criteria

	Hot Side	Cold Side	Units
Media	Water	Seawater	
Volumetric Rate	19,511/1.231	19,511/1.231	GPM / m^3/s
Mass Flow Rate	162,727/1230	167,215/1265	lb./min / kg/s
T_{in}	54.7/12.6	43.7/6.5	°F / °C
T_{out}	45.3/7.4	52.9/11.6	°F / °C
Density	8.34/999	8.57/1027.6	lb./gal / kg/m^3
LMTD 1.7°F / 0.95°C Load 86,677 x 10^3 BTU/hr / 2.54 x 10^4 kW			

The coefficient of heat transfer or operating K value for titanium heat exchanger can be expected to be about 7166.4 $W/m^2°C$ or 1262 $J/ft^2.h.°F$.

Using Equation (30), with the load of 2.52 x 10^4 kW,

$$2.52 \times 10^4 \text{ kW} = 7166.4 \text{ W/m}^2\text{°C} \cdot 0.95\text{°C} \cdot \text{Area}$$

ields 3702 m^2 for the effective surface area of the heat exchanger. is is larger than any previously known heat exchangers for this purpe, but based on information on other "off the shelf" models, a sin- unit of this capacity might occupy 66 cubic meters of space in facility. It is unlikely that a single unit would be used. It ght be wise to have several heat exchangers and excess capacity so t one could be shut down for cleaning and maintenance without loss air conditioning capacity. On the other hand, since the entire air

nditioning system is intended to be backed up by a conventional sys-
n, minimum redundancy in the heat exchangers is required.

Even without redundancy, space, weight and transportation con-
raints might restrict the size. Therefore, it is assumed here that
ree units would be installed in parallel, each of effective area
44 m^2 and possibly containing as many as 1,000 plates. This is only
estimate of what would be required; the units vary from manufac-
rer to manufacturer.

PUMPS

The design of the pumping system includes an onshore sump for
e pumps. The sump may be wet or dry. The onshore sump would provide
:ess for maintenance. The construction of the sump could be adja-
nt to the Ala Wai Canal, and only a short trench or tunnel would be
quired.

The pumps themselves could be three or four centrifugal pumps,
anged in parallel, to provide the required discharge head of 58.63
ers. Variable speed pumps could be used, and controls could be set
in a manner which allows only one or many pumps to be used on the
water side at one time. The flow rate could be controlled automat-
lly by a temperature sensor on the discharge side of the heat ex-
nger. This would reduce power consumption during off peak times.

The pressure drop in the heat exchanger is of great importance.
ater area required by the heat exchangers increases the pressure

op which occurs and increases the pumping power required to force
h the seawater on the cold side and the fresh water on the air con-
ioning side through the heat exchanger at the desired flow rates.
e power required for this task can be expected to be several times
gher than the power required to pump the cold water to the site.
y 93 kW out of a projected 720 kW is required for seawater deliv-
r. Most of the remaining 617 kW power requirement is for the head
ss in the heat exchangers. The heat exchangers could be expected to
esent a pressure loss of as much as 83 KPA or 12 psi.

GENERAL APPROACH

The intent of this design is that a conventional air condition- system be installed as both a back-up system and for contingency ling in the event the complex requires supplementation during peak d periods, or due to unforeseen future facility expansion. There- e, the capital cost of the proposed system is the same as a conven- nal system plus the added expense of the seawater loop. The total litional capital costs analyzed are only those associated with the struction of the large heat exchangers, the cold water supply and charge pipe, the related pumps and control system as shown in Ta- (5). Some of the assumptions made, such as the inclusion of a con- tional back-up air conditioning system, are extremely servative. In subsequent design iterations, such overkill would bably be eliminated in the name of cost cutting.

STIMATE OF CAPITAL COSTS

Costs of pumps and controls are difficult to estimate without de- led mechanical drawings. The estimate of $750 K for materials and 0 K for labor, equipment and installation is a rough guess based estimates of other similar projects. The accepted method of esti- ing cost of the pipeline material is by weight. A cost of $1.00 pound of HDPE is used. This price has been fairly stable over the several years, although recently, due to an "oil glut," the es have dipped as low as $.80/lb. According to the vendors, how-

er, this is not expected to last. The cost of $431 per meter of
pe is arrived at by using the pipe's diameter of 1200 mm and thick-
ss of 57 mm to calculate the volume of material in 1 meter of pipe,
en multiplying that by density of 956 kg/m^3 and the cost of $1/lb.
2.21 lbs./kg. The assumption made earlier was that the pipe would
of uniform thickness throughout the cold water loop. The nearshore
ction of 2600 meters length and offshore section of 4600 meters
ngth are then multiplied by the cost per meter of $431.

Table (5) Capital Costs of Seawater Air Conditioning System (all costs are loaded, including overhead and profit)

Item Description		Material Cost			Construction Cost (Labor & Equipment) (thousands $)	Total Cost (thousands $)
		Unit Cost (thousands $)	Quantity	Total Material Cost (thousands $)		
Pumps and Controls			LS	750	500	1,250
Seawater Supply Pipe	Nearshore Sections	0.431/m	2,600	1,121	4,480	5,601
	Offshore Sections	0.431/m	4600	1,983	2,974.5	4,957.5
ffluent Pipe (all nearshore) edundant effort with supply pipe)		0.431/m	2,600	1,121	1,121	2,242
Heat Exchanger		450	3	1,350	250	1,600
& Misc. Plumbing		100	LS	100	included	100
Subtotals				6,425	9,325.5	15,750
Contingency 10%				643	933	1,570
Permits/EIS		?	?	?	?	?
Totals				7,067	10,258	17,325.5

The construction costs (labor and equipment) for the pipeline e more difficult to estimate. According to Makai Ocean Engineering Waimanalo, Hawaii, their experience in designing the pipelines at ahole Point has been that the overall cost of the pipelines are gen- ally five times the cost of the material for the pipeline. There- re, labor and equipment costs can be estimated to be about four mes the material price (Van Ryzin, Leraand 1991). Other estimates the Keahole Point project indicate that the cost of the offshore ction may be a little lower since there is no excavation involved. e multiplication factor may therefore be reduced to 3.5 for the off- ore section (Syed, Nihous and Vega 1991).

In this case the multiplication factor is lowered further for veral reasons, most of which relate to economy of scale. Previous timates were based on much shorter pipelines. The cost of setting a staging and assembly area, building forms for pre-cast concrete chors and bringing a large tugboat in from the mainland would all examples of large one time set-up costs. Spreading these costs er the length of the offshore section of the pipeline could signifi- ntly reduce the cost. Therefore the total multiplication factor ed is 2.5 in this estimate.

The figures in Table (5) for construction costs for the two sec- ons of supply pipe are arrived at by using multiplication factors f 5 and 2.5 for the nearshore and offshore, respectively. The multi- l.cation factor for the effluent pipe is reduced to 2, instead of 5,

ause of much duplication of effort in the nearshore zone for the luent and supply pipelines.

The cost of the heat exchanger is generally 25% of the cost of overall system (Van Ryzin 1991). Using that as a basis, the heat hanger and related installation might be expected to cost about 5 M. However, these estimates were based on systems with large on-re distribution systems and supply pipes less than one third the gth of the one in this design.

Based on 1991 costs quoted on smaller titanium heat exchangers, heat exchanger requirements could be met at a cost of approxi-ely $450,000 each, for three units including freight and taxes. An ditional $100,000 is added to this for miscellaneous valves and mbing materials. Labor and equipment costs for installation and ting could run another $250,000. The total heat exchanger esti-e, including miscellaneous plumbing, is $1.7 M. It should be noted e that the use of an aluminum alloy heat exchanger would reduce cost of this item considerably. Such a cost cutting option would ost certainly be considered in subsequent design iterations.

Adding all these components yields a sum of just over ,750,000, as shown in Table (5). A 10% contingency cost is added reach a total capital cost of $17,325,500 for the total seawater ion of the system.

As stated earlier, the air conditioning system for the conven- complex will also include a complete conventional air condition-

system for reasons already discussed. The $17.3M in capital costs in addition to the cost of that conventional system.

OPERATING COSTS

Although it is expected that the actual maintenance costs for seawater air conditiong system will be less than for a conventional system, for this conservative analysis, it is assumed that the intenance costs over the life of the system will be roughly the ne as that for a conventional system. The largest difference is exted to come from the energy savings. These costs could be reduced between 70% and 90%.

ENERGY ANALYSIS

In general, one ton of cooling capacity requires about 1.0 kW of ctricity, all of which would be derived from the electrical power d in the conventional air conditioning system. This energy is used drive the circulation pumps, fans and compressors (or chillers). chillers are the big energy consumers. In a typical air condition- system, 800-900 kW of electricity is used per 1000 tons for the rigeration alone. This is predicted to be replaced by 75-150 kW 1000 tons of cooling for the seawater system (Van Ryzin, Leraand 2).

The calculation done herein only includes the energy required to come the head loss due to friction in the cold water supply pipe. is not considered is the head loss due to the resistance of the

t exchangers. Also, the friction loss of the effluent pipe would o have to be considered. It may, therefore, appear reasonable to ept the 75-100 kW per 1000 tons, since the projects described by Ryzin and Leraand (1992) involved pumping the water to ultimate tinations several meters above sea level, thereby increasing the d loss.

Using an estimate of 100 kW per 1000 tons of capacity, the re- t is 720 kW. The energy savings can then be computed as follows:

850 kW/1000 tons - 100 kW/1000 tons = 750 kW/1000 tons

750 kW/1000 tons x 7.2 (1000 tons) = 5400 kW

This is the reduction in energy used when the system is running full capacity. For the purpose of analysis, 4500 equivalent full d hours is used for Honolulu. This is multiplied by the savings full load to get the total annual energy savings in kilowatt- rs,

$$5400 \text{ kW} \times 4500 = 24.3 \times 10^6 \text{ kWh}$$

Multiplying the annual savings in kilowatt-hours by the electric e of about $.10/kWh, the total annual savings to the convention ter operator is calculated to be $2.43 M.

ASH FLOW ANALYSIS

With an initial capital cost of $17.3 M and an annual savings if .3M, the payback period can be determined based on the net present le of two options. The first option is constructing the seawater

conditioning system as described herein. The second option is the nothing" option which would mean installing only the conventional conditioning system, and not realizing the annual energy savings not spending $17.3 M in capital costs up front.

A rough evaluation by the Payback Method (White et al 1977) indi-es that the payback period, based on a zero interest rate, would

$$\frac{\$17.3 \times 10^6}{2.43 \times 10^{6\$/yr}} = 7.12 \text{ years}$$

An estimate of payback taking interest and inflation into ac-nt could assume an interest rate of 8% and an inflation rate of The combined effects could be assumed to be 5%. Using the com-nding tables in the back of any economics book, a value for the ber of periods of compounding can be applied. In this case, = 7.12. In the "i = 5%" table, n is between 7 and 8 periods. In-polation yields 7.55 years.

Both of the previous methods are fairly inexact. A more detailed lysis using 8% as the discount rate and 3% as a projected infla-n rate can also be used. The analysis used by Syed et al (1991) l the following two formulas:

$$PW = \frac{A(1)}{(1+d)} \sum_{j=1}^{j=n} \left[\frac{(1+i)}{(1+d)} \right]^{j-1} \qquad \text{Eq. (33)}$$

where:

PW = present worth of the revenue stream

(savings in this case)

i = inflation rate

d = discount rate

c = capital investment

A(j) = annual net revenue at the end of n years

= for any year j

n = the number of years (or periods of time)

When $d \neq i$, the geometric series can be written as,

$$S = A(1) \cdot \frac{\left[1 - \left[\frac{(1+i)}{(1+d)} \right]^n \right]}{(d-i)} \qquad \text{Eq. (34)}$$

The payback period is the time N of value n at which S is equal the capital cost, C. Therefore,

$$N = \frac{\log \left[\frac{1 - C(d-i)}{A} \right]}{\log \left[\frac{(1+i)}{(1+d)} \right]} \qquad \text{Eq. (35)}$$

In this particular case of the seawater air conditioning system,

$$N = \frac{\log \left[\frac{1 - 17.3(.08 - .03)}{2.43} \right]}{\log \left[\frac{(1+.03)}{(1+.08)} \right]} = 9.28 \text{ years}$$

This payback period is based on parameters such as the interest t: and inflation rate, which could easily change. Currently, the in- r·st rate is closer to 7%. Using this in a similar calculation u d yield a payback period of 8.80 years. Since the interest rates

r the next several years appear to be more likely to rise a lit-
, the more conservative estimate of a 9.28 year payback based on
8% discount rate is favored here.

There are other variables as well. Not only is the capital in-
tment estimate rough, but there is no experience in maintaining a
water air conditioning system of this size, and therefore, the as-
ption that maintenance costs for the conventional versus seawater
conditioning would be the same could also be a source of error.

A 9.28 year payback period may not be acceptable to the State,
ever, comparisons of payback periods, although a popular method,
not necessarily the best method of comparing the options. More re-
nsible methods of comparing engineering options include comparing
present or future value of the resulting cash flows over the life
ectancy.

The life expectancy is different for the different major compo-
s of the system. The supply pipeline is designed to last 50 years
n the standpoint of strength. The heat exchangers, however, are ex-
ed to last only 10 to 20 years. An analysis taking these factors
account could compare the initial capital cost of $17.3 M to the
rn of $2.43 M per year, adjusted for inflation over a period of
ears. In the pessimistic case, the heat exchangers could be re-
ed at a present cost of $1.7 M at the 10 year mark. The compari-
of present valued would be calculated as follows. Assuming the
tric rates increase at the 3% rate of inflation, the savings of

43 M per year in 1994 dollars over 20 years (a geometric series) equal to $2.43 M (P/A, i, j, n), where i = 8%, j = 4%, n = 20 rs. This is equal to $2.43 M x 61.7459 = $150 M. The present worth the $1.7 M heat exchangers replacement in 10 years is still $1.7 M.

Therefore, the total present worth over a 20 year period is esti- ed to be:

-$ 17.3 M initial capital costs

+$150.0 M present worth of electricity savings over 20 years

-$ 1.7 M possible replacement of heat exchangers in 10 years

$131.0 M

Therefore, the present worth of the seawater air conditioning tem can be estimated to be $131 M greater than the comparable con- tional system.

Summary and Conclusions

Previous research has clearly shown that use of seawater air con-
ioning is technically feasible. The projects at Keahole Point and
er locations have demonstrated that such a system can be designed,
lt and operated successfully.

The use of seawater air conditioning for Waikiki can be applied
the convention center project or perhaps other large building de-
opments in Hawaii. The use of an HDPE pipeline of 1200 mm diameter
ld supply enough cold water to accomplish this.

The Ala Wai canal could provide a suitable conduit for a cold
er supply pipeline in the case of the convention center, although
roject of this magnitude could be expected to draw as much atten-
n as the controversial convention center itself.

The problems of installing a pipeline through the nearshore area
be overcome with current technology and sound design procedures.
t exchanger technology is also currently available "off the
lf," which could accomplish the desired goals of air conditioning
onvention center size project.

Conditions at Waikiki require a cold water supply pipe to extend
east 5600 meters offshore, to reach a minimum depth of 525 me-
s. The flow through a pipe of this length would not experience ex-
sive temperature gains if the flow were fast enough and the pipe
f large enough diameter. This contributes to the argument that a
er system is more feasible.

No adverse environmental effects are foreseen from the use of e cold water, and fouling of either the cold water supply pipe or e heat exchangers is also not expected to be a problem. The dredg- g operations off Waikiki could be an obstacle from the environ- ıtal standpoint, however, these types of operations are carried out equently throughout Hawaii and the world, and procedures for mini- zing environmental impact are standard.

To avoid much of the environmental impact, directional drilling other tunneling techniques could be employed in the nearshore ıe, however, at a cost of %500 to $1000 per foot for a tunnel of e size being considered, the economics are prohibative.

Overall, the use of seawater air conditioning for the convention ıter is estimated herein to cost approximately $17.3 M more than a ıventional system. Offsetting the cost is an estimated energy sav- gs of $2.43 M per year. This is calculated to provide a 7 to 9 year back period, and a net savings of many tens of millions over the e of the system. This is the major incentive for constructing a tem of this type.

The estimates and calculations used to arrive at these conclu- ıns were preliminary and admittedly rough. As more building details ome available, it is recommended each of the areas of the design refined, and the cost benefits of a seawater air conditioning sys- for the Waikiki Convention Center be examined in more detail.

Appendix A-1

PIPE FLOW CALCULATIONS

Units:

1L kg ≡ 1M sec ≡ 1T degC ≡ 1Q

$$= 1 \cdot kg \cdot \frac{m^2}{sec^3} \qquad J := 1 \cdot kg \cdot \frac{m^2}{sec^2}$$

Parameters:

water density:

$$= 1027.6 \cdot \frac{kg}{m^3}$$ For Seawater at 6 deg Celsius

$$= 9.81 \cdot \frac{m}{sec^2}$$

water viscocity (kinematic):

$$= 1.56 \cdot 10^{-6} \cdot \frac{m^2}{sec}$$ For Seawater at 6 deg Celsius

pipe wall thickness is:

$= .057 \cdot m$

ide and outside pipe diameters are:

$:= 1.086 \cdot m$ $D_o := D + 2 \cdot t$ $D_o = 1.2$ length

l Pipe length:

$:= 7200 \cdot m$

ired Volumetric flow:

$$: 1.231 \cdot \frac{m^3}{sec}$$

oss sectional area of pipe (inside):

$$:= \pi \cdot \frac{D^2}{4} \qquad A = 0.926\ \text{length}^2$$

ow velocity inside pipe:

$$= \frac{Q}{A} \qquad V = 1.329\ \text{length}\ \text{time}^{-1}$$

nolds number for pipe flow:

$$:= \frac{V \cdot D}{v} \qquad Re_d = 9.252\ 10^5$$

ce the Reynolds Number is much grater than 2300, The flow is turbulent. In s case a friction factor must be determined from empirical tables. Since a yethelene pipe will be used, the smooth pipe assumption can be made.

m Appendix B-2 (Fox & McDonald,1973) the friction factor is:

= .0167

pressure change and head loss can then be calculated as follows:

$$= \frac{V^2}{2} \cdot \left[f \cdot \frac{L}{D} \right] \cdot \rho \qquad \delta P = 1.005\ 10^5\ \text{mass}\ \text{length}^{-1}\ \text{time}^{-2}$$

l loss:

$$:= \frac{\delta P}{\rho} \cdot \frac{1}{g} \qquad h_l = 9.966\ \text{length}$$

Appendix A-2

gram to determine stresses on pipe due to hydrostatic pressure:

e Material Charactoristics:

erial: HDPE

sity: $\rho_p := 965 Kg/cu.meter$

son's Ratio: $u := .3$

p modulus (50 yr):

$$E_{50} := \frac{200 \cdot 10^6}{.300} \quad N/sq.meter$$

Demensions:

= 1.200 meters

$= \frac{OD}{2}$ $R_o = 0.6$ meters

= 21

$= \frac{OD}{SDR}$ $t = 0.057$ meters

: $OD - 2 \cdot t$

$ID = 1.086$ meters

$= \frac{ID}{2}$ $R_i = 0.543$ meters

)er unit lenth:

$= \rho_p \cdot \pi \cdot \left[R_o^2 - R_i^2 \right] \cdot 1$ Mass = 197.984 kg

ccording to another criteria, the critical buckling
trength can be calculated as follows:

ckling Strength Coefficient: $C_B := .5$

rrection factor for deflection: $C_D := 1.0$

me dependent pipe stiffeness:

$$PV_V := 53.7 \cdot E_{50} \cdot \frac{I}{ID^3}$$

fety Factor: $FS := 2$

en the critical buckling pressure is given by:

$$P_{cr} := C_B \cdot C_D \cdot \frac{[PV_V]^{.5}}{FS}$$

$$P_{cr} = 164.877 \quad PSI$$

SI units:

$$SIP_{cr} := P_{cr} \cdot 6895$$

$$SIP_{cr} = 1.137 \cdot 10^{6} \quad N/sq.meter$$

oment of Inertia:

$$:= \frac{t^3}{12} \qquad I = 1.555\ 10^{-5}\ \text{meters}^4$$

ength: L := 7200 meters

low Calculations:

stimated A/C Load: LOAD := 7200 tons

nversion factor for tons to Kilowatts: CONV := 3.5 KW/ton

ad In KW: LOADKW := LOAD·CONV

$$LOADKW = 2.52\ 10^{4}\ \text{Kilowatts}$$

ecific Heat of Seawater: $C_p := 3.985$ KJ/Kg.deg C

nsity of Seawater: $\rho := 1027.6$ Kg/cu.meter

scosity of Seawater: (inematic) $v := 1.56 \cdot 10^{-6}$ sq.meter/sec

pected Seawater temperature change in Heat Exchanger: $\delta T := 5$ degC

quired Mass flow:

$$M := \frac{LOADKW}{C_p \cdot \delta T} \qquad M = 1.265 \cdot 10^{3}\ \text{Kg/sec}$$

quired Volumetric Flow:

$$Q := \frac{M}{\rho} \qquad Q = 1.231\ \text{cu.M/sec}$$

overted to Gallons per minute:

Qgal := Q·1000·.2642·60

$$Qgal = 1.951\ 10^{4}\ \text{Gal/min}$$

lw Velocity:

$$V := \frac{Q}{\pi \cdot ID^2 \cdot .25} \qquad V = 1.329\ \text{M/sec}$$

nolds #:

$$Re := \frac{V \cdot ID}{v} \qquad Re = 9.252\ 10^{5}$$

culation of Pressure loss, δP:

er Appendices B-1 and B-2, where f=friction factor:

$$f := .0167$$

$$\delta P := .5 \cdot V^{2} \cdot f \cdot \frac{L}{ID} \cdot \rho \qquad \delta P = 1.006\ 10^{5} \quad N/sq.meter$$

$$\delta psi := \frac{\delta P}{6895} \qquad \delta psi = 14.585 \quad psi$$

Loss:

$$h_{l} := \frac{\delta P}{\rho} \qquad h_{l} = 97.863 \quad meters$$

ulate Collapse strength of pipe:

criteria for Hydrostatic collapse strength of round pipe is as follows:

$$p_{c} := \frac{24 \cdot E_{50} \cdot I}{1 - u^{2}} \cdot \frac{1}{\left[\frac{OD + ID}{2}\right]^{3}}$$

$$p_{c} = 1.832\ 10^{5} \quad N/sq.Meter$$

$$pPSI_{c} := \frac{p_{c}}{6895}$$

$$pPSI_{c} = 26.563 \quad PSI$$

CALCULATION OF TEMPERATURE CHANGE OF WATER IN PIPE

sumptions: 1. Flow over the outside of the pipe is constant.
2. Cp is constant for seawater in the pipe

fine Variables:

pe section length and average outside temperature:

$s := 1200 \cdot m$ $T_o := 7.5 \cdot degC$

$= 0.926\ length^2$

ow Temperature at the beginning of the pipe section:

$:= 6.0 \cdot degC$

$:= 0.3$ $n_2 := 0.4$

beflow velocity:

$= 1.329\ length\ time^{-1}$

$= 1.028\ 10^{3}\ mass\ length^{-3}$

$k_p := 1.51 \cdot \frac{W}{m \cdot degC}$

$1.56\ 10^{-6}\ length^{2}\ time^{-1}$

rntdl number:

r:= 11.00

yamic Viscosity of seawater:

$= \rho \cdot v$

$0.002\ mass\ length^{-1}\ time^{-1}$

imeter based Reynolds Number for pipe flow:

$e = 9.252\ 10^{5}$

ne temperature of the water outside the pipe is referred to as To. This
nanges as a function of depth as shown in fig (8) in the text and the depth
lso changes as a function of the length with respect of the pipe. Therefore
ne temperature is an unknown function of both depth and length of pipe. To
alculate the heat loss over the length of the pipe, an average temperature
ıtside the pipe could be used. A better solution can be arrived at by
ssuming an average temperature over various sections of the pipe and
alculating the heat loss and temperature change for each section. The heat
oss and temperature change in each section can then be added. This is the
ethod employed.

ne calculation for the one section is as follows:

or heating...

$$n := n_2$$

lculating the Nusselt number using Reynolds and Prandtl numbers for fully
veloped turbulent flow in smooth tubes:

$$Nu_d := 0.023 \cdot Re_d^{0.8} \cdot Pr^{n} \qquad Nu_d = 3.558 \times 10^{3}$$

e heat transfer coeficient for the inside surface of the pipe is then
lculated as follows:

$$h_i := \frac{k_p \cdot Nu_d}{D_i} \qquad h_i = 4.948 \times 10^{3}\ \text{mass}\ \text{time}^{-3}\ \text{temperature}^{-1}$$

le thermal resistance for the inside surface of the pipe is :

$$R_i := \frac{1}{h_i \cdot \pi \cdot D_i} \qquad R_i = 5.924 \times 10^{-5}\ \text{mass}^{-1}\ \text{length}^{-1}\ \text{time}^{3}\ \text{temperature}$$

c the unit length of polyethylene pipe the thermal resistance is calculated
s follows, where the "r's" denote the inside and outside radii of the pipe
rl the k is the thermal conductivity of the pipe material.
p

$$D_i := \frac{D_o}{2} \qquad r_o := \frac{D_o}{2} \qquad k_p = 1.51\ \text{mass}\ \text{length}\ \text{time}^{-3}\ \text{temperature}^{-1}$$

$$\text{:=} \frac{\ln\left[\frac{r_o}{r_i}\right]}{2\cdot\pi\cdot k_p} \qquad R_p = 0.011\ \text{mass}^{-1}\ \text{length}^{-1}\ \text{time}^{3}\ \text{temperature}$$

r calculating the the heat transfer coeficient for the outside surface of
e pipe, a flow velocity must be assumed and a Reynolds number calculated in
der to choose an applicable emperical relationship.

locity of flow over the outer surface of the pipe:

$$\text{:=} 1.0\cdot\frac{m}{sec}$$

ynolds number for flow over outer surface of the pipe:

$$e_f := V_f\cdot\frac{D_o}{v} \qquad Re_f = 7.692\ 10^{5}$$

le Nessalt number is then:

$$\text{:=} 0.023\cdot Re_f^{.8}\cdot Pr^{.4}$$

$$= 3.07\ 10^{3}$$

herefore the heat transfer coefficient for the outside of the pipe can be
simated to be:

$$\text{:=} k_p\cdot\frac{Nu_f}{D} \qquad h_o = 4.269\ 10^{3}\ \text{mass}\ \text{time}^{-3}\ \text{temperature}^{-1}$$

ased on the outer surface area of the pipe:

$$\text{:=} \pi\cdot D_o\cdot L_s \qquad A_o = 4.524\ 10^{3}\ \text{length}^{2}$$

$$R_o := \frac{1}{h_o \cdot A_o} \qquad R_o = 5.179\ 10^{-8}\ \text{mass}^{-1}\ \text{length}^{-2}\ \text{time}^{3}\ \text{temperature}$$

The overall heat transfer coeficient for the pipe and seawater is :

$$U_o := \frac{1}{\frac{A_o}{A_i} \cdot \frac{1}{h_i} + A_o \cdot \frac{\ln\left[\frac{r_o}{r_i}\right]}{2 \cdot \pi \cdot k_p \cdot L_s} + \frac{1}{h_o}}$$

$$U_o = 0.974\ \text{mass}^{-3}\ \text{time}^{-1}\ \text{temperature}^{-1}$$

$$q := U_o \cdot A_o \cdot \left[T_o - \left[T_{x1} + \frac{\delta T}{2}\right]\right] \quad \square$$

Calculation of the change in temperature within the pipe from the beginning of the section of interest to the end of that section.

The specific heat for seawater varies with temperature and salinity. For the purpose of this calculation the value from Appendix B-3 for 30‰ salinity and 0 deg Celsius is close enough.

Specific heat of seawater:

$$C_p := 4013.6 \cdot \frac{J}{kg \cdot degC}$$

Mass flow is calculated:

$$M := \rho \cdot Q$$

$$M = 1.265\ 10^{3}\ \text{mass time}^{-1}$$

$$q := M \cdot C_p \cdot \delta T \quad \square$$

o solve for the root, an initial estimate for δT is:

$$T := 0.1 \cdot degC$$

$$\delta T := root\left[M \cdot C_p \cdot \delta T - U_o \cdot A_o \cdot \left[T_o - \left[T_{x1} + \frac{\delta T}{2}\right]\right], \delta T\right]$$

$\delta T = 0.001$ temperature

nitial water temperature in the next section of pipe to be evaluated must be alculated by adding this temperature change.

$$T_{x2} := T_{x1} + \delta T \qquad T_{x2} = 6.001 \text{ temperature}$$

his procedure is repeated for each section of pipe.

Pipe Section Number	Length (meters)	Avg. Depth (meters)	T_0 (avg.) °C	T_1 (entrance) °C	ΔT °C	T_2 (end) °C
1	1200	450	7.5	6.000	8.67×10^{-4}	6.001
2	1800	325	11.0	6.001	.004	6.005
3	400	175	19.5	6.005	.011	6.016
4	800	60	26.0	6.016	.017	6.033
5	1200	10	26.5	6.033	.018	6.051
6	1200	2.5	27.0	6.051	.018	6.069
Totals	6600				.069	

Appendix A-4

following is a sample calculation designed to demonstrate how forces due offshore currents and incoming waves can affect a submerged pipeline. The ameters used are modeled after a pipeline sitting directly on a flat floor but they could be altered easily to represent other situations. The gram is writtten in Mathcad version 2.5. This format allows the formulas be easily read and expainations of each step inserted."
is intended that the waves be transformed for shoaling, refraction and fraction before use for actual design work. The Army Corps of ineers Shore Protection Manual or published programs for that specific pose would allow refraction and diffration to be considered.
After determining the wave and current induced forces, the program executes analysis of the free body diagram of a unit section of pipe. Parameters pipe size and material can be altered to fit a given situation. The last of the program predicts the amount of force required to stabilize the under the given wave and current conditions. The assumtion is made that stabilizing forces would result from adding weight in the form of casting pipe in concrete. The required force is translated into equivalent ht and volume of concrete.
convenience, all the necessary parameters can be entered from the tom of the program so that the results can be viewed on the same screen and crolling through the program is required.

9 = the wave period of interest in seconds

9.81 = gravity in metric units

60 = local water depth in meters

15 = wave hieght of interest in meters

first step is to calcIate the deep water wavelength, L_o

$$L_o := \frac{g}{2\pi} T^2$$

$$L_o = 126.466 \quad \text{meters}$$

iterative process is required to calculate the local wavelenth. To
complish this the program requires an initial estimate."

= 30 can be used as an initial estimate.

calculation of wavelenth is as follows:"

$$L := \text{root}\left[\frac{d}{L_o} - \frac{d}{L}\tanh\left[2\pi\frac{d}{L}\right], L\right]$$

$$L = 125.826 \quad \text{meters}$$

wave number is calculated to simplify the calculation of the wave heights.

$$k := 2\frac{\pi}{L}$$

calculation of the local wave height resulting from the deep water wave
ident on the local region could then be calculated for an in-depth
lysis of wave conditions in a given area. However, for the purpose of this
culation, the local wave height is assumed to be H meters.

H = 15 meters

following values can be computed and compared to those found in table C-1
the Shore Protection Manual.

$$\frac{d}{L_o} = 0.474 \qquad \frac{d}{L} = 0.477$$

test the local wave height for breaking.

$$H := \text{if}\left[\frac{H}{d} > 0.78, H, 0\right]$$

ocal) is H when H/d > .78, otherwise assume it is 0

mary of Wave Information Results:

= 15 meters T = 9 seconds d = 60 meters

$$\frac{H}{d} = 0.25$$

CULATION OF PARTICLE MOTIONS:

ulation of water particle velocities and accelerations is as follows
e $\omega := 2\frac{\pi}{T}$ and Ω is the phase angle of the wave, evaluated at the
om. $z := -d$ Several values of Ω are examined so that the maximum
es can be determined. Since the 'worst case' expected is for the
ction of wave propogation to be at a 60 degree angle of incidence, α or
epipe, the velocities and accelerations in the following equations include
factor cos α.

1.047 = radians angle of incidence

= 0,15 ..180 = phase angle in degrees

$= j \frac{\pi}{180}$ = phase angle in radians

$$= \frac{H\ g\ k}{2\ \omega}\left[\frac{\cosh(k\ (d + z))}{\cosh(k\ d)}\right]\sin\left[\Omega_j\right]\sin(\alpha)$$

dt=a

$$= \frac{-(H\ g\ k)}{2}\frac{\cosh(k\ (d + z))}{\cosh(k\ d)}\cos\left[\Omega_j\right]\sin(\alpha)$$

j	
0	degrees
15	
30	
45	
60	
75	
90	
105	
120	
135	
150	
165	
180	

CULATION OF WAVE INDUCED FORCES:

calculate the forces due to the particle velocities resulting from wave
ion the following equations are employed. First however, the coeficients
irag, lift and added mass must be assumed.

= 0.5 C_L = 1.4 C_m = 3.3

r characteristics of the pipe geometry and fluid density are also
ired.

= 1.1 meters = approximate outside diameter

^{3}
=1.027 1kg/cu.meter = density of seawater

equation for the horrisontal force, Fx is the Morrison's Equation.

$$F_{x_j} := C_m \rho \pi \left[\frac{D_o}{2}\right]^2 a_j + \frac{1}{2}\rho D_o C_d u_j^2$$

the verticle direction per unit length, the following equation is used.

$$F_{z_j} := C_L \frac{\rho}{2} D_o u_j^2$$

F_{x_j} Newtons	F_{z_j} Newtons
-1.022 10	10.94
-983.097	40.828
-870.342	81.656
-693.374	122.484
-467.167	152.372
-210.048	163.312
58.326	152.372
318.885	122.484
554.655	81.656
751.7	40.828
899.505	10.94
990.911	0
$1.022\ 10^3$	

CES RESULTING FROM STEADY CURRENT: (Alone)

re is also a drag force and lift force associated with the steady current
dent on the pipeline. The same equations are used to calculate the forces
the x and z-directions with the velocity of the current used instead of
particle velocities. Note that there is no acceleration since the current
steady' and therefore the inertial portion of the Morrison's equation is
l to zero.

to knot current, U = 1m/sec, is assumed to be present in the longshore
rction. This is assumed to be in a perpendicular direction to the
rction of the pipe.

=1 m/s = steady current velocity

$$Fx_{current} := \frac{1}{2} \rho D_o C_d U^2$$

$$Fz_{current} := C_L \frac{\rho}{2} D_o U^2$$

$= 282.425$ NEWTONS $Fz_{current} = 790.79$ NEWTONS

ıng the components for the x and z-directions for both the wave
ıced motion and the current velocities doesn't necessarily yield a total
ːe of the water on the pipeline per unit area. To examine the total
ıltant forces on the pipelinse for a given section under given wave
liions, the partical velocities and accelerations from the waves and
ːents must be added to get an effective velocity, U_e and acceleration, a_e.
ı the combined particle motions can be used in the Morrison's and lift
ıtion.

$$U_{e_j} := U + u_j$$

$$a_{e_j} := a_j$$ Since the current is assumed to be steady.

indices on the wave forces remain from the previous calculations
ːuse the phase angle at which the 'worst' case for the total resultant
ːes on the pipe is not yet known.

$$F_{x_j} := C_m \rho \pi \left[\frac{D_o}{2}\right]^2 a_{e_j} + \frac{1}{2} \rho D_o C_d U_{e_j}^2$$

$$F_{z_j} := C_L \frac{\rho}{2} D_o U_{e_j}^2$$

the total forces at various wave phase angles determined, only the forces the weight of the pipe, bouyancy of the pipe and friction with the bottom in to be determined for an analysis to be complete. The bouyancy and weight of the pipe are constant, however the friction force resisting ment in the x-direction is dependant on the normal force of the pipe on bottom. The normal force is the sum of all the vertical forces and varies the wave phase angle.

bouyant, F_b, force per unit length on the pipeline is calculated as ows.

$$F_b := \rho \, g \, \pi \, \frac{D_o^2}{4}$$

where: ρ = density of seawater

D_o = pipe diameter (outside)

$F_b = 9.574 \, 10^3$ Newtons/unit length

eight of the pipe per unit length is the sum of the weight of the pipe ial itself plus the weight of the water contained. The density of the material, ρ_p, and the inside diameter, D_i, of the pipe are required for calculation.

1 meter $\rho_p = 954$ kg/cu.meter

$$W_p := \rho \, g \, \pi \cdot \frac{D_i^2}{4} + \rho_p \, g \, \pi \, \frac{D_o^2 - D_i^2}{4}$$

$W_p = 9.456 \cdot 10^3$ Newtons/unit length

inspection, it can be seen that added weight is required in order to come the bouyant force so that the pipe will stay on the seafloor. To rmine the downward force per unit length required to accomplish this, the of all the verticle forces, Fz, must be determined.

$Fz_{T_j} := F_{z_j} + F_{b_j} - W_p$

Fz_{T_j}

908.904 Newtons
$1.106 \cdot 10^3$
$1.309 \cdot 10^3$
$1.499 \cdot 10^3$
$1.654 \cdot 10^3$
$1.756 \cdot 10^3$
$1.791 \cdot 10^3$
$1.756 \cdot 10^3$
$1.654 \cdot 10^3$
$1.499 \cdot 10^3$
$1.309 \cdot 10^3$
$1.106 \cdot 10^3$
908.904

orce required to overcome bouyancy throughout the wave cycle is the um value of F_z.

$F_z := \max\left[Fz_T\right]$

$F_z = 1.791 \cdot 10^3$ Newtons/unit length

volume concrete, V_c, required to weight the pipe is calculated as follows.

weight of the required concrete in seawater water is equal to F_z.

$:= F_z$

$= 2.4\ 10^3$ Kg/cu.meter

crete

$$V_c := \frac{W_c}{\left[\rho_{concrete} - \rho\right] g}$$

= 0.133 cubic meters of concrete/meter length of pipe

amount of concrete is the minimum required to overcome the bouyant and
t case lift force for water motion. This may not be enough to provide the
al force on the seafloor required for friction to overcome the maximum
force in the x-direction. To accomplish this, the weight of the concrete
ired is calculated for each phase angle and the maximum is taken. This is
because the the maximum lift affects induces a minimum normal force but
will not occur at the phase angle at which the maximum drag occurs.
sum of the forces in the x-direction, Fx_T, is calculated using an assumed
ficient of friction, f, of 0.5 applied to the normal forces.
).5

$$Fx_{T_j} := F_{x_j} - f\left[Fz_{T_j} - W_c\right]$$

aximum of the sum computed above is the resultant force which must be
ome by adding wieght to induce a greater normal force. The amount of
ion normal force required is equal to the resultant maximum resultant
, F_x, devided by the coeficient of friction. Again this normal force
e applied by adding concrete weights, W_{ac}, to the pipe.

$$W_{ac} := \frac{\max\left[Fx_T\right]}{f}$$

$$W_{ac} = 3.491\ 10^{3}$$

total downward force required by concrete attached to the pipe is the sum
he concrete added to overcome the lift and bouyancy and that added to
required to overcome drag.

$$:= W_{ac} + W_{c}$$

$$= 5.281\ 10^{3}$$

volume of concrete is computed in a similar manner as previously.

$$V_T := \frac{F_T}{\left[\rho_{concrete} - \rho\right] g}$$

$$V_T = 0.392 \quad \text{cubic meters per unit length}$$

nd, this amount of concrete weighs:

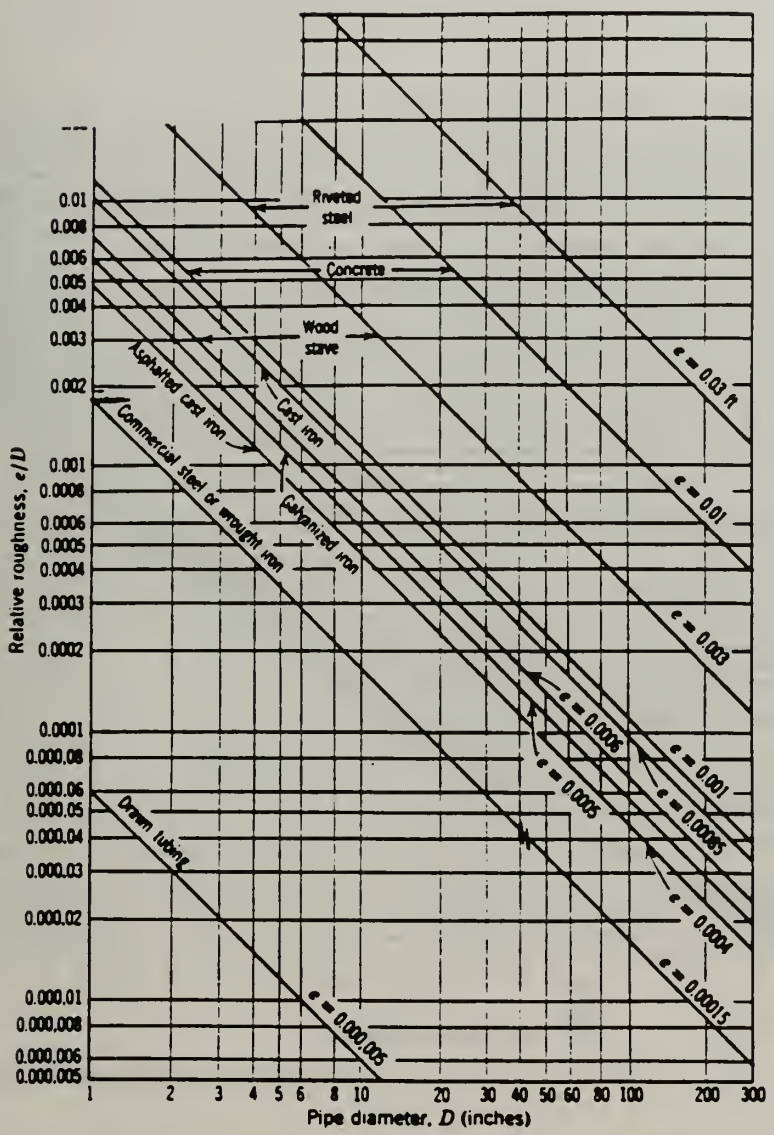
$$W_T := V_T \rho_{concrete} g$$

$$W_T = 9.232\ 10^{3} \quad \text{Newtons}$$

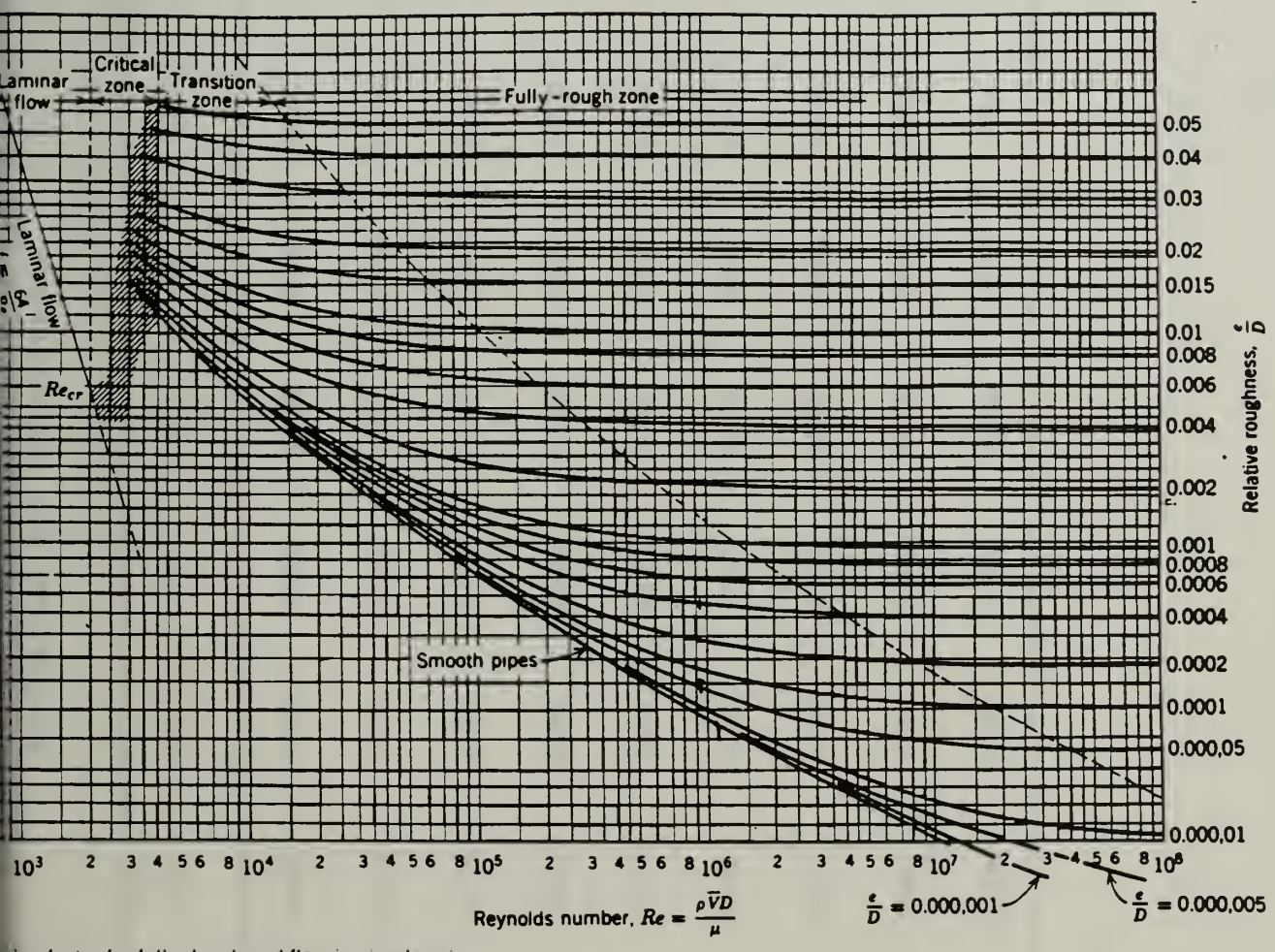
Sample Results

Water Depth (meters)	Wave Height, H_o (meters)	Period, T (seconds)	Min. Vertical Applied Force Required per Meter Length of Pipe for Stability (Newtons)
200	9	9	1476
200	9	10	1485
200	9	12	1574
200	9	15	2016
200	15	9	1477
200	15	10	1492
200	15	12	1641
200	15	15	2392

Appendix B-1



Relative roughness values for pipes of common engineering materials



ion factor for fully-developed flow in circular pipes

Specific Heat of Seawater at Constant Pressure ($Jg^{-1}°C^{-1}$) at Various Salinities and Temperatures

Salinity, 0/00	0°C	5°C	10°C	15°C	20°C	25°C	30°C	35°C	40°C
0	4.2174	4.2019	4.1919	4.1855	4.1816	4.1793	4.1782	4.1779	4.1783
5	4.1812	4.1679	4.1599	4.1553	4.1526	4.1513	4.1510	4.1511	4.1515
10	4.1466	4.1354	4.1292	4.1263	4.1247	4.1242	4.1248	4.1252	4.1256
15	4.1130	4.1038	4.0994	4.0982	4.0975	4.0977	4.0992	4.0999	4.1003
20	4.0804	4.0730	4.0702	4.0706	4.0709	4.0717	4.0740	4.0751	4.0754
25	4.0484	4.0428	4.0417	4.0437	4.0448	4.0462	4.0494	4.0508	4.0509
30	4.0172	4.0132	4.0136	4.0172	4.0190	4.0210	4.0251	4.0268	4.0268
35	3.9865	3.9842	3.9861	3.9912	3.9937	3.9962	4.0011	4.0031	4.0030
40	3.9564	3.9556	3.9590	3.9655	3.9688	3.9718	3.9775	3.9797	3.9795

From Riley and Skirrow, 1975

Mechanical, Thermal and Processing Properties of Glass Fiber Reinforced Plastics

Material		Specific Gravity	Thermal Conductivity W/(m K)	Specific Heat, J/(kg K)b	Tensile Strength MPa	Tensile Modulus, GPa	Flexural Modulus, GPa	Compressive Strength, MPa	Impact Strength Izod at 22oC, J/m
Glass fiber-reinforced thermoplastics (RTP)	polypropylene	1.04	14.5		45	3.7	3.6	172	59
	poly(phenylene sulfide)	1.64	3.47	1.05	152	14.1	13.1	145	80
	acrylonitrile-butadiene-styrene terpolymer (ABS)	1.22	2.42		76	6.2	6.0	97	64
	poly(phenylene oxide) (PPO)	1.21	6.57	.84 - 1.67	100	6.3	5.2	121	96
	poly(ethylene terephthalate)	1.56	11.2		145	9.0	8.6	172	96
Unreinforced thermoplastics (TP)	poly carbonate	1.20	2.34	1.26	66	2.3	2.3	86	854
	polypropylene	0.89	2.10	1.88	34	0.7	0.9-1.4	24	50-1000
	poly(phenylene sulfide)	1.30	2.89		66	3.3	3.8	110	<27

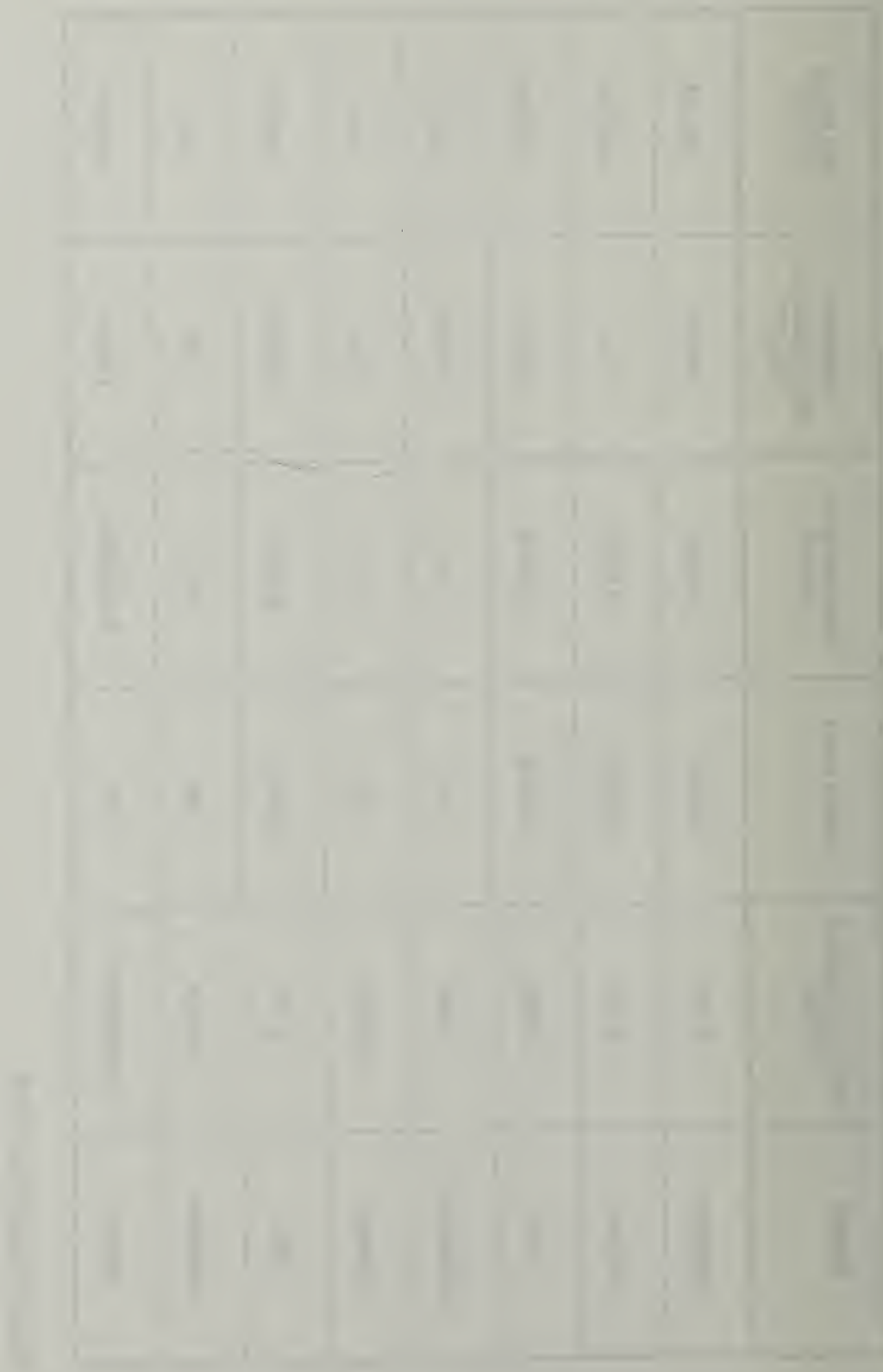
	styrene terpolymer (ABS)								
	poly(phenylene oxide) (PPO)	1.10	1.59	0.84-1.67	54	2.6	2.3-2.8	83	270
	styrene-acrylonitrile (SAN)	1.05	1.21	1.38	66	2.8	3.8	97	16-24
	poly(butylene terephthalate)	1.31	1.76-2.89		57	1.9	2.3-2.8	59	43
	poly(ethylene terephthalate)	1.34	1.51	1.42	59	2.8	2.4-3.1	76	13-35
	ASTM A-606 HSLA steel, cold rolled	7.75	43.3	0.46	448	207		448	
	SAE 1008 low carbon steel, cold rolled	7.86	60.6	0.42	331	207		331	
Metals	AISI 304 stainless steel	8.03	16.3	0.50	552	193		552	
	TA 2036 aluminum, wrought	2.74	159	0.88	338	70		338	
	ASTM B85 aluminum, die cast	2.82	91.8		331	71		331	

ASTM AG40A zinc, die cast	6.59	113	0.42	283	75		283

From Rosato and Rosato, 1990

Material	Modulus of elasticity E, GN/m^2	Poisson's ratio, v	Shear modulus G, GN/m^2	Coefficient of linear expansion α, 10-6 / °C	Mass Density 10^3 kg / m^3
Aluminum	68-78.6	0.32-0.34	25.5-26.5	20.0-24.1	2.66-2.88
Iron, cast	89-145	0.21-0.30	35.8-56.5	10.4	6.95-7.34
Steel	193-220	0.26-0.29	75.8-82.0	9.9-12.8	7.72-7.86
Stainless Steel	193-207	0.30	73.1	14.9-16.9	7.64-7.92
Titanium	106-114	0.34	41.4	8.82	4.51
Glass	50-79	0.21-0.27	26.2-32.4	5.94-9.54	2.38-3.88
Polyethylene	0.14-0.38	0.45	0.117	180	0.91
Rubber	0.00076-0.0041	0.50	0.0003-0.001	126-198	1.0-1.24

From Crandall et al 1978



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